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Smithson et al.

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(54) **CONTINUOUSLY VARIABLE TRANSMISSION**

(71) Applicant: **Fallbrook Intellectual Property Company LLC**, San Diego, CA (US)

(72) Inventors: **Robert A Smithson**, Cedar Park, TX (US); **Brad P Pohl**, Leander, TX (US); **Oronde J Armstrong**, Austin, TX (US); **Donald C Miller**, Fallbrook, CA (US); **Daniel J Dawe**, Austin, TX (US); **Fernand A Thomassy**, Liberty Hill, TX (US); **Mathew P Simister**, Austin, TX (US); **Wesley R Poth**, Leander, TX (US); **Jon M Nichols**, Georgetown, TX (US); **Charles B Lohr**, Austin, TX (US)

(73) Assignee: **Fallbrook Intellectual Property Company LLC**, Cedar Park, TX (US)

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(58) **Field of Classification Search**
USPC 476/36–38, 40, 42
See application file for complete search history.

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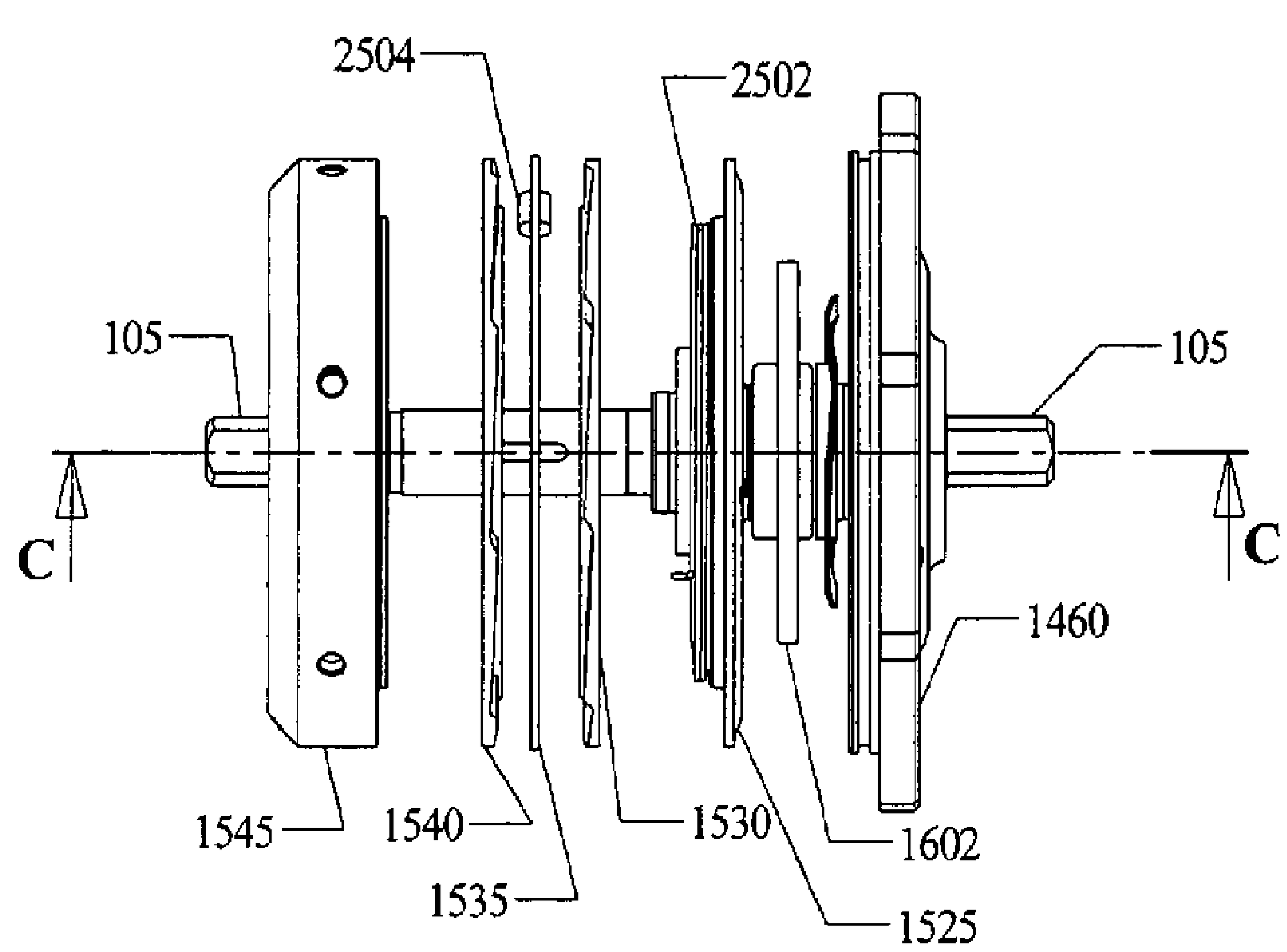
Primary Examiner — William C Joyce
(74) *Attorney, Agent, or Firm* — Knobbe Martens Olson & Bear LLP

(57) **ABSTRACT**

A continuously variable transmission (CVT) having a main shaft configured to support and position various components of the CVT. Shift cam discs cooperate with ball-leg assemblies to shift the transmission ration of the CVT. Load cam discs, a torsion disc, rolling elements, and a hub cap shell are configured to generate axial force, transmit torque, and manage reaction forces. In one embodiment, a splined input shaft and a torsion disc having a splined bore cooperate to input torque into the variator of the CVT. Among other things, various ball axles, axle-ball combinations, and reaction force grounding configurations are disclosed. In one embodiment, a CVT having axial force generation means at both the input and output elements is disclosed.

17 Claims, 26 Drawing Sheets



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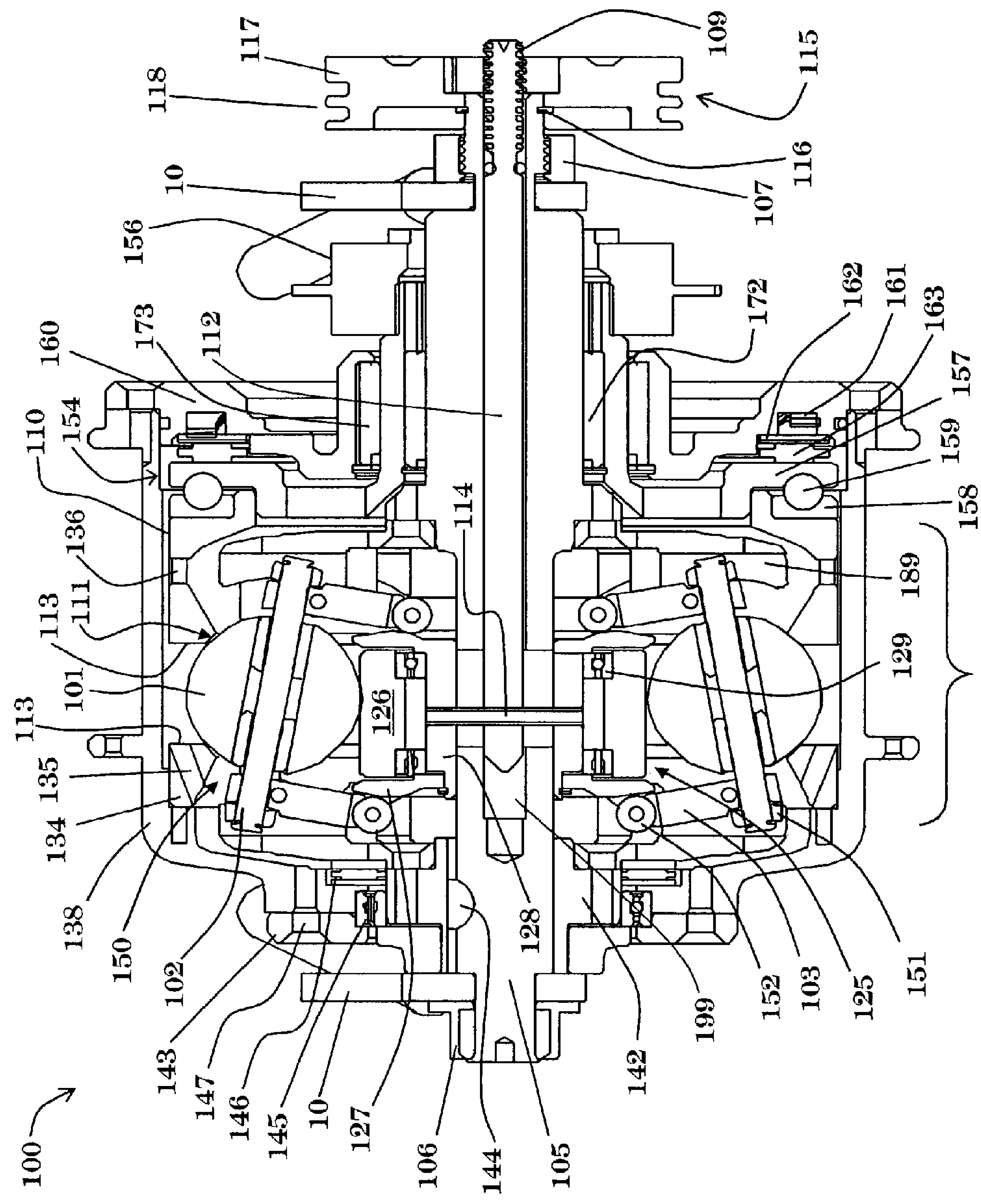
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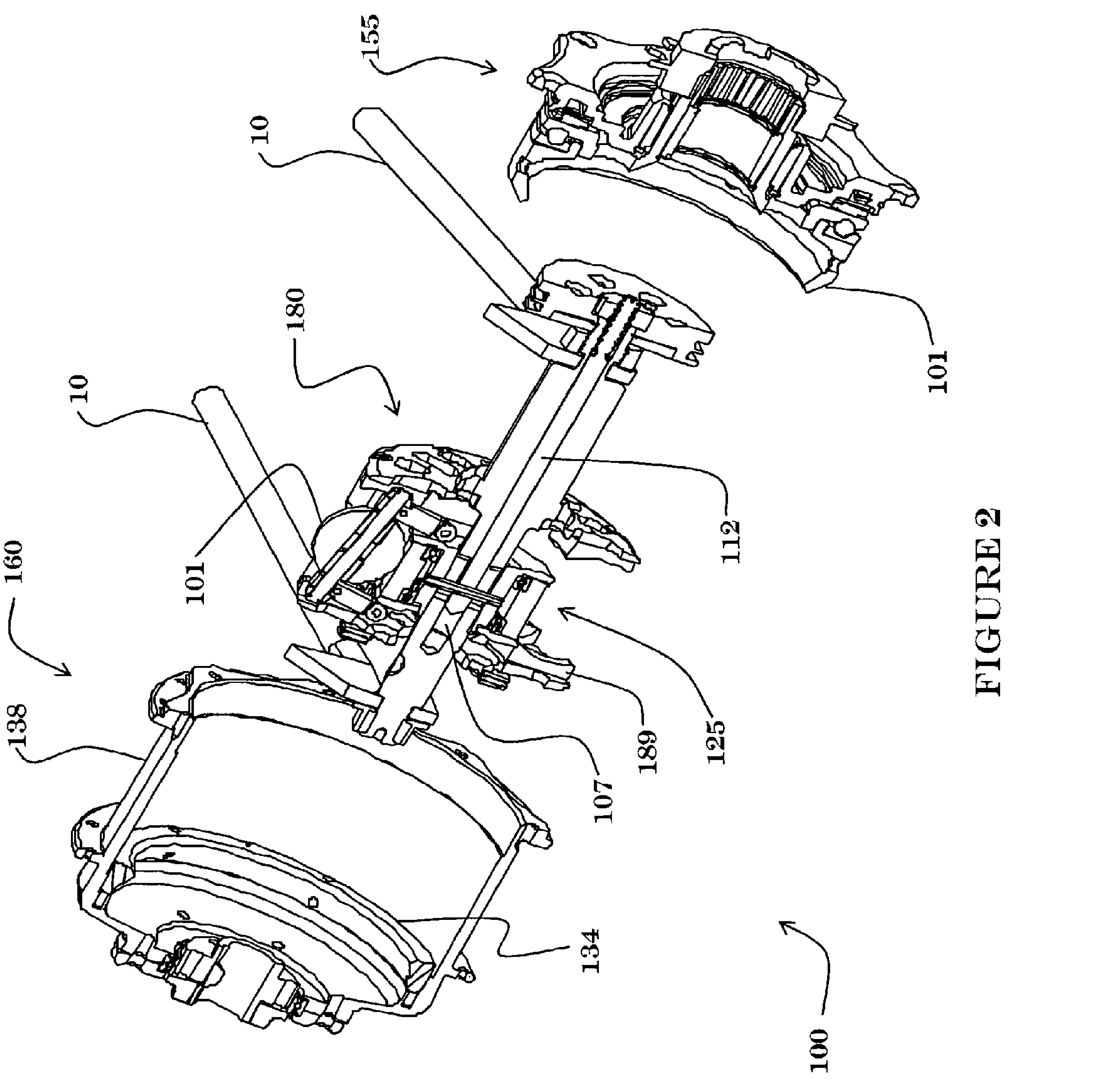
FIGURE 1



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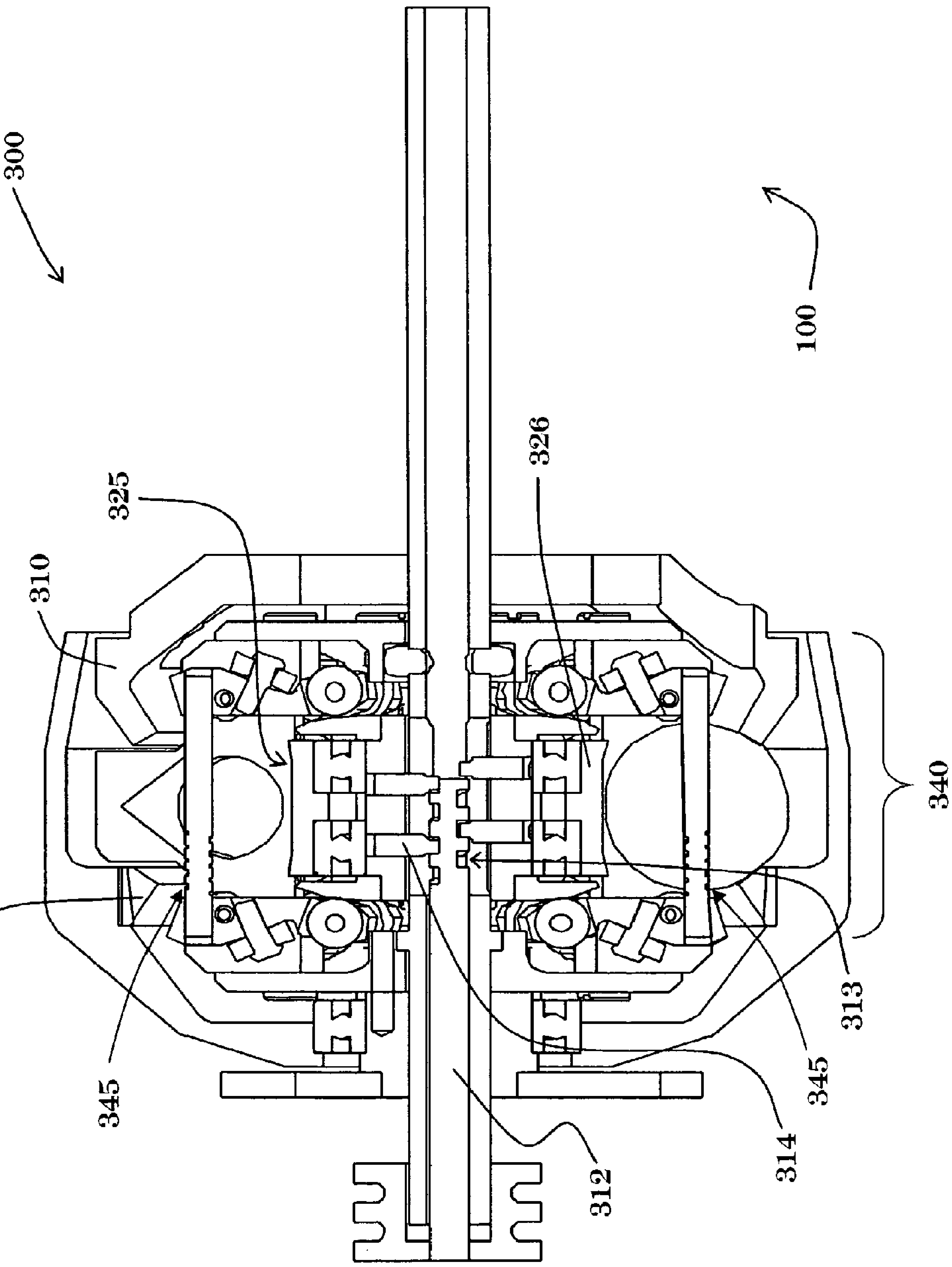

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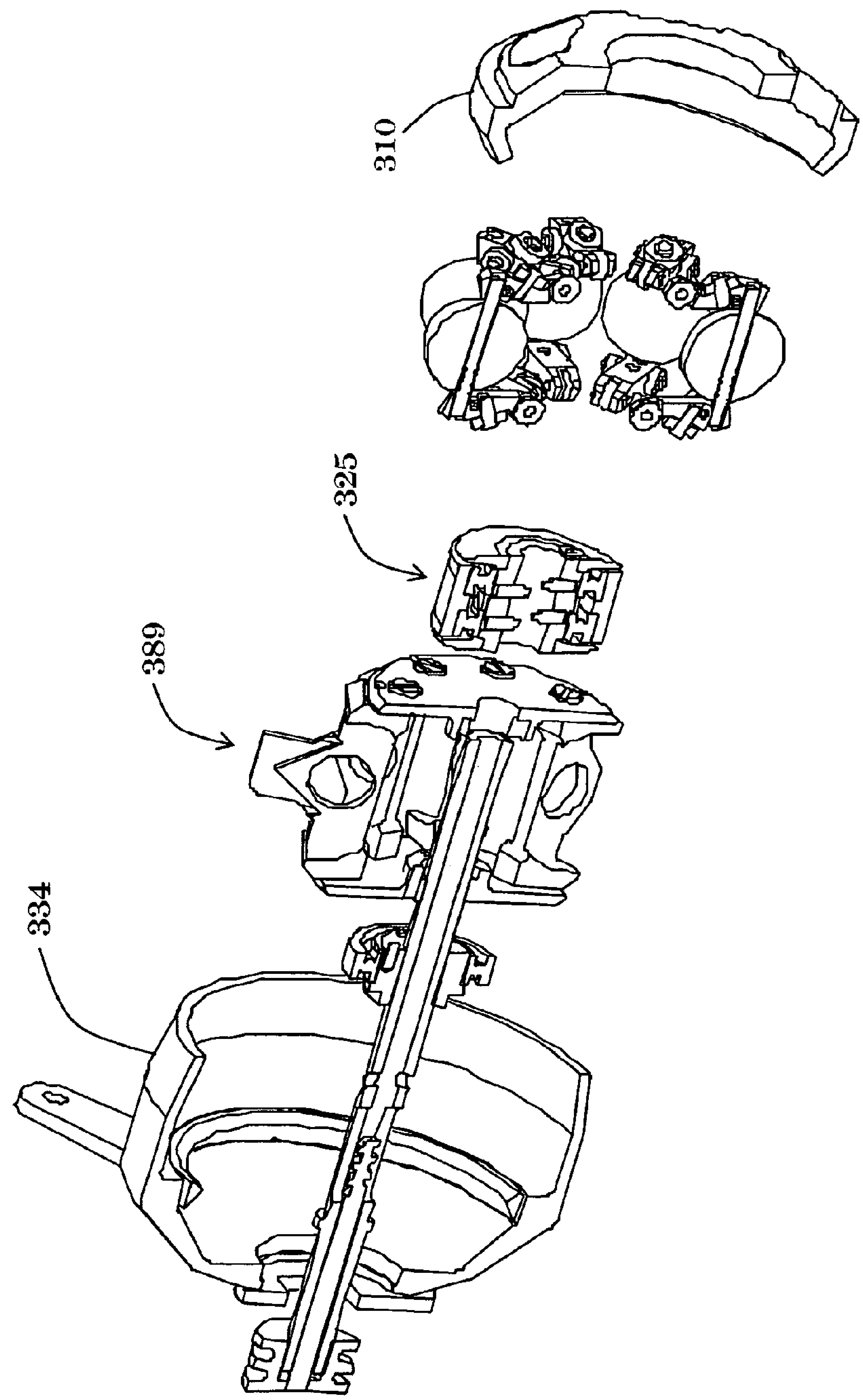

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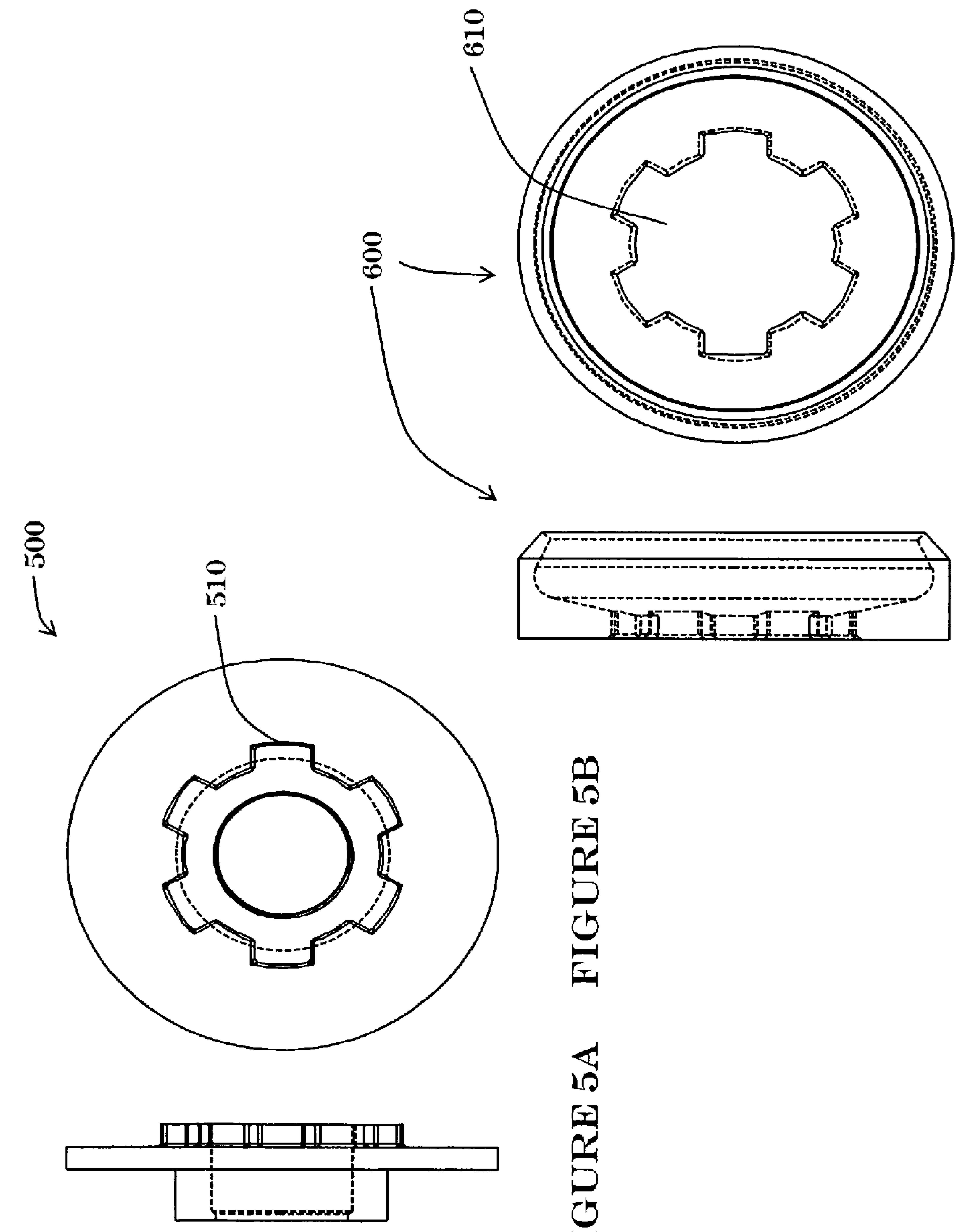
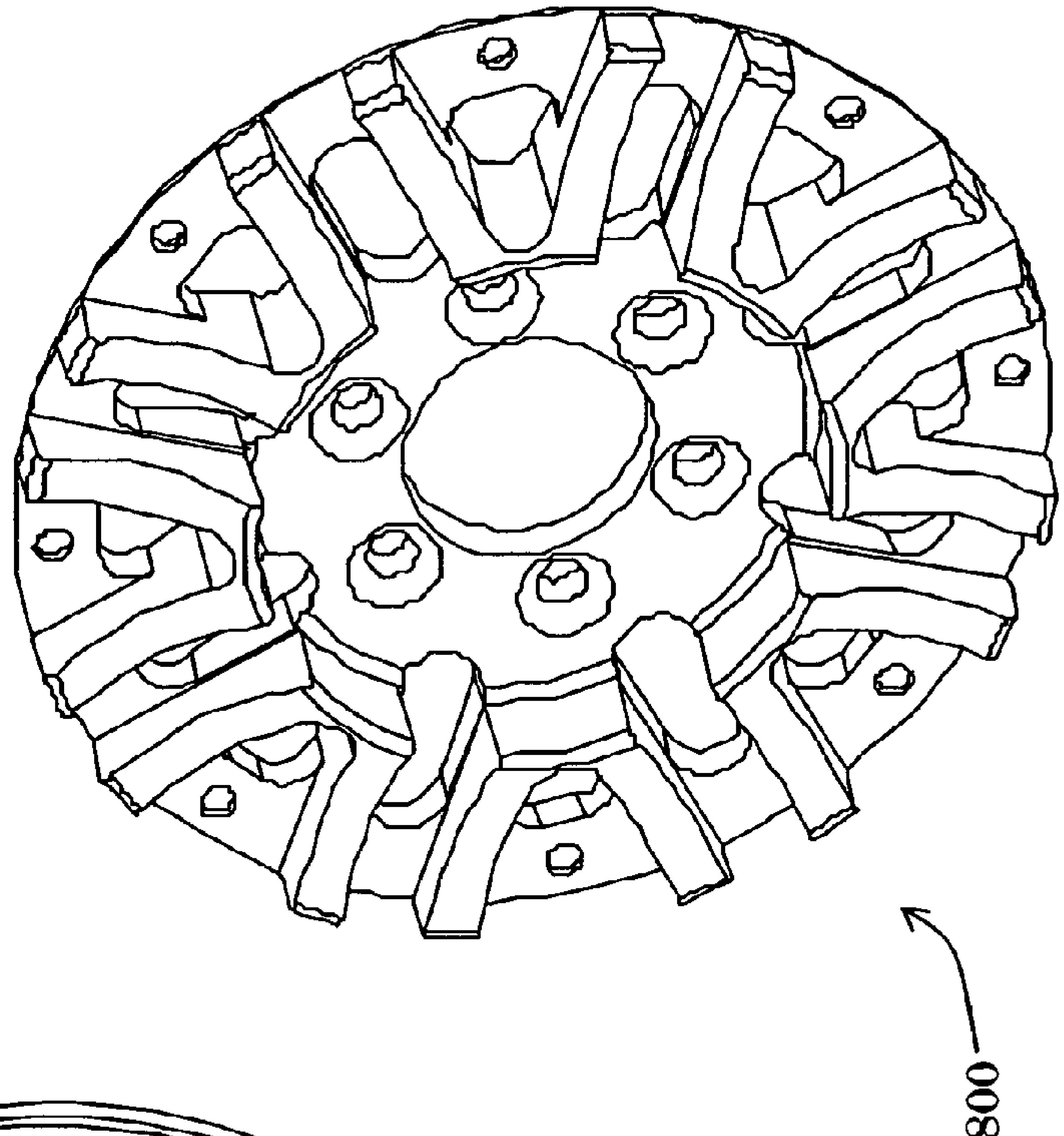


FIGURE 5A

FIGURE 5B

FIGURE 6A

FIGURE 6B



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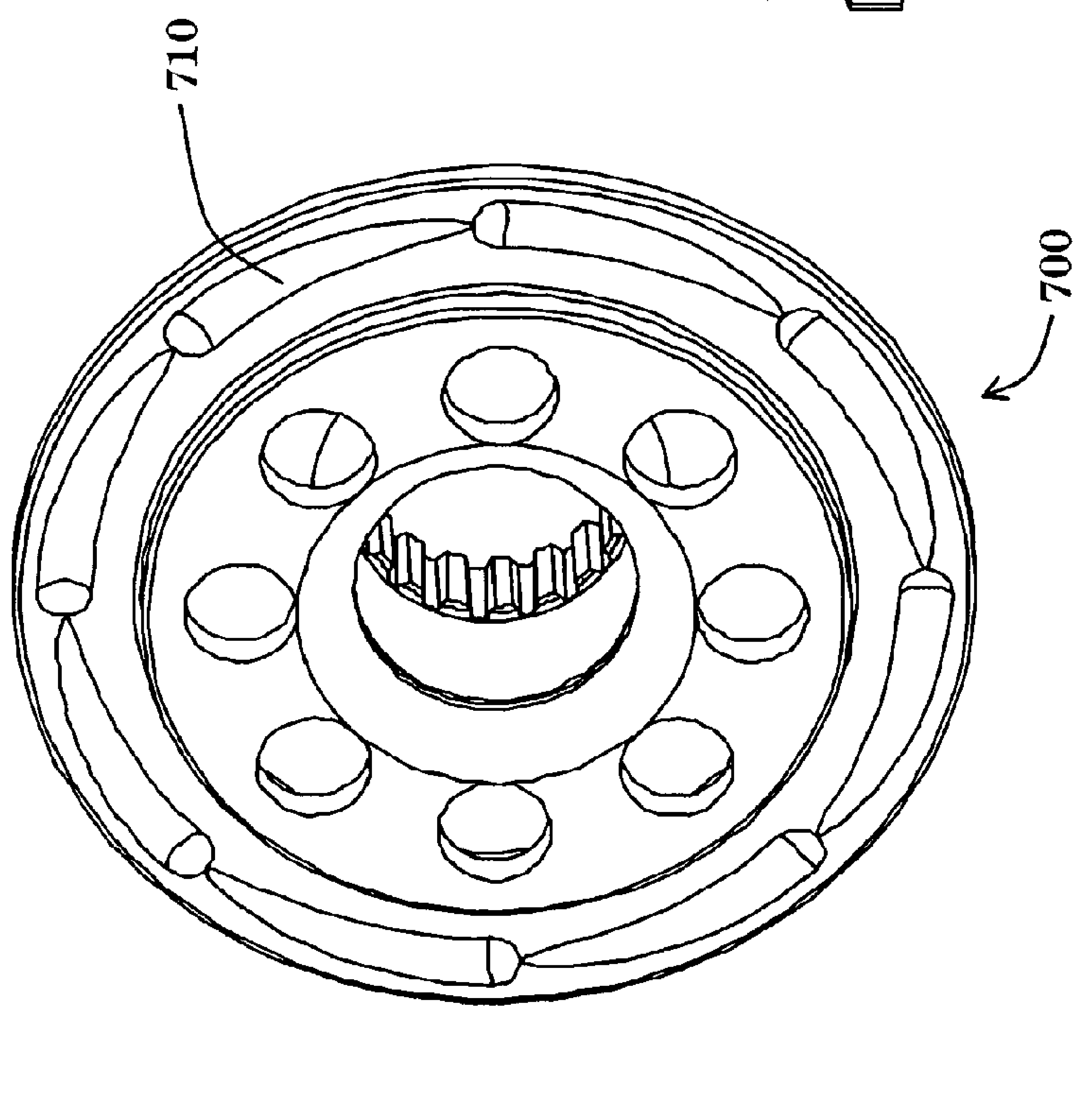

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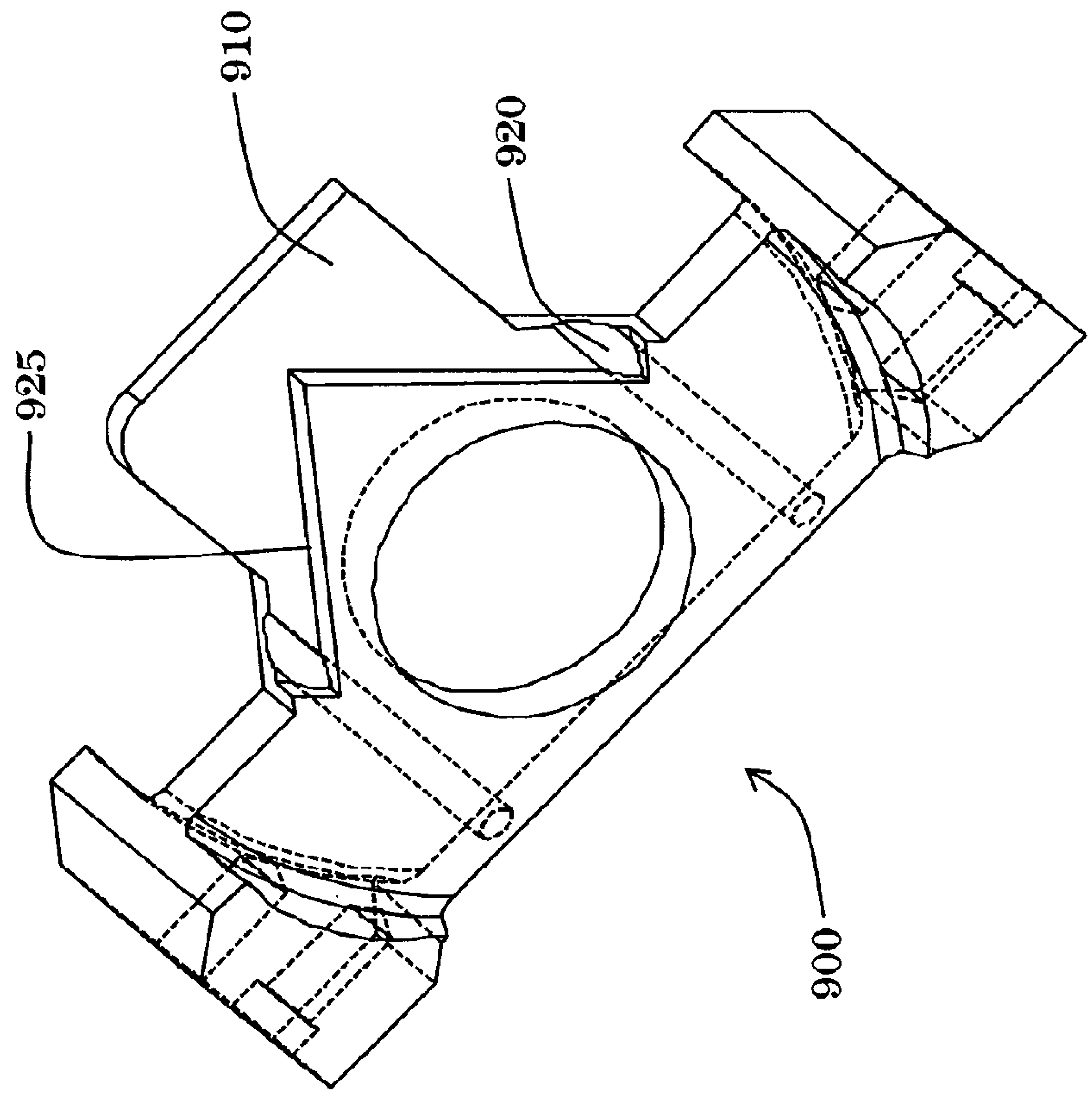

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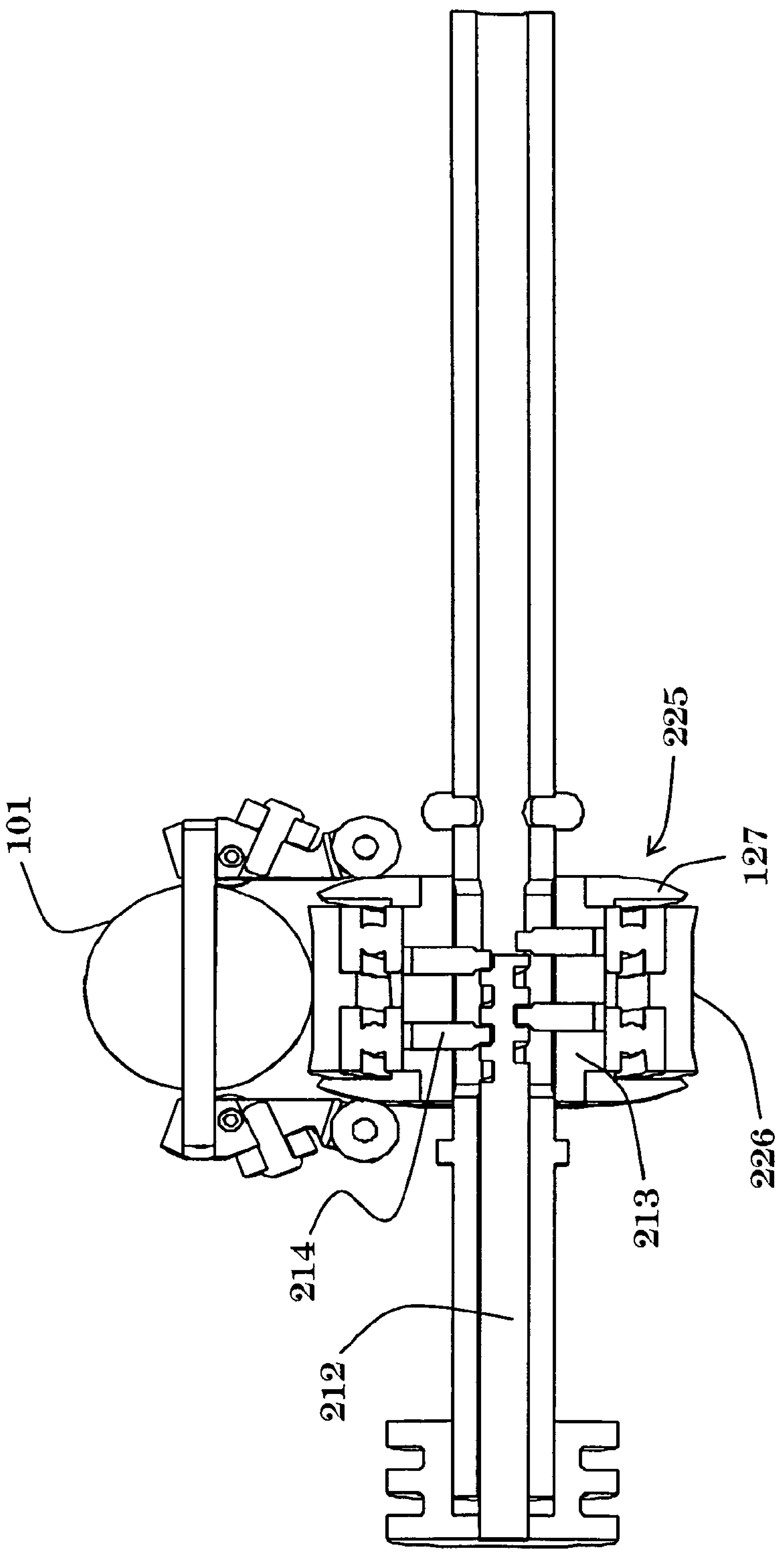

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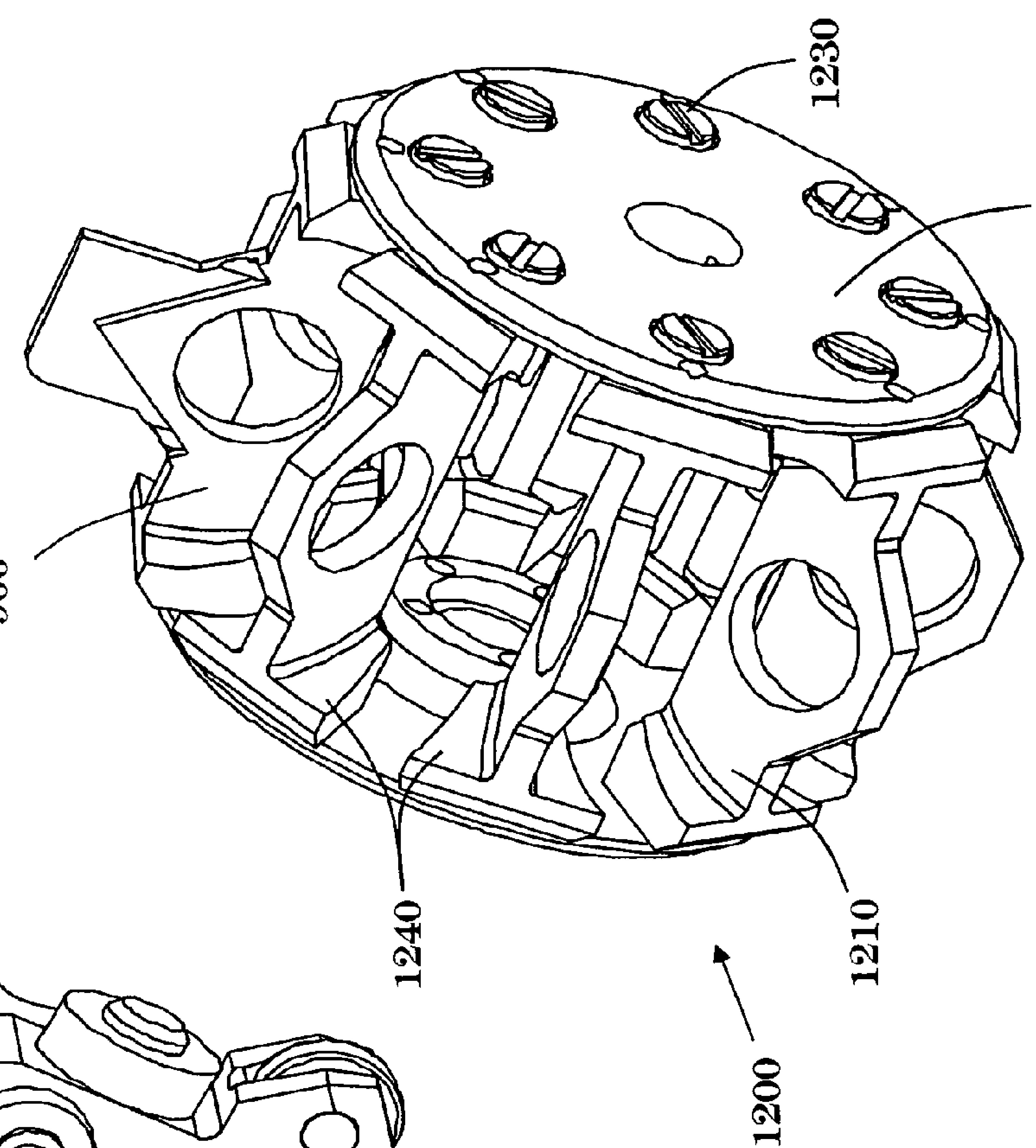

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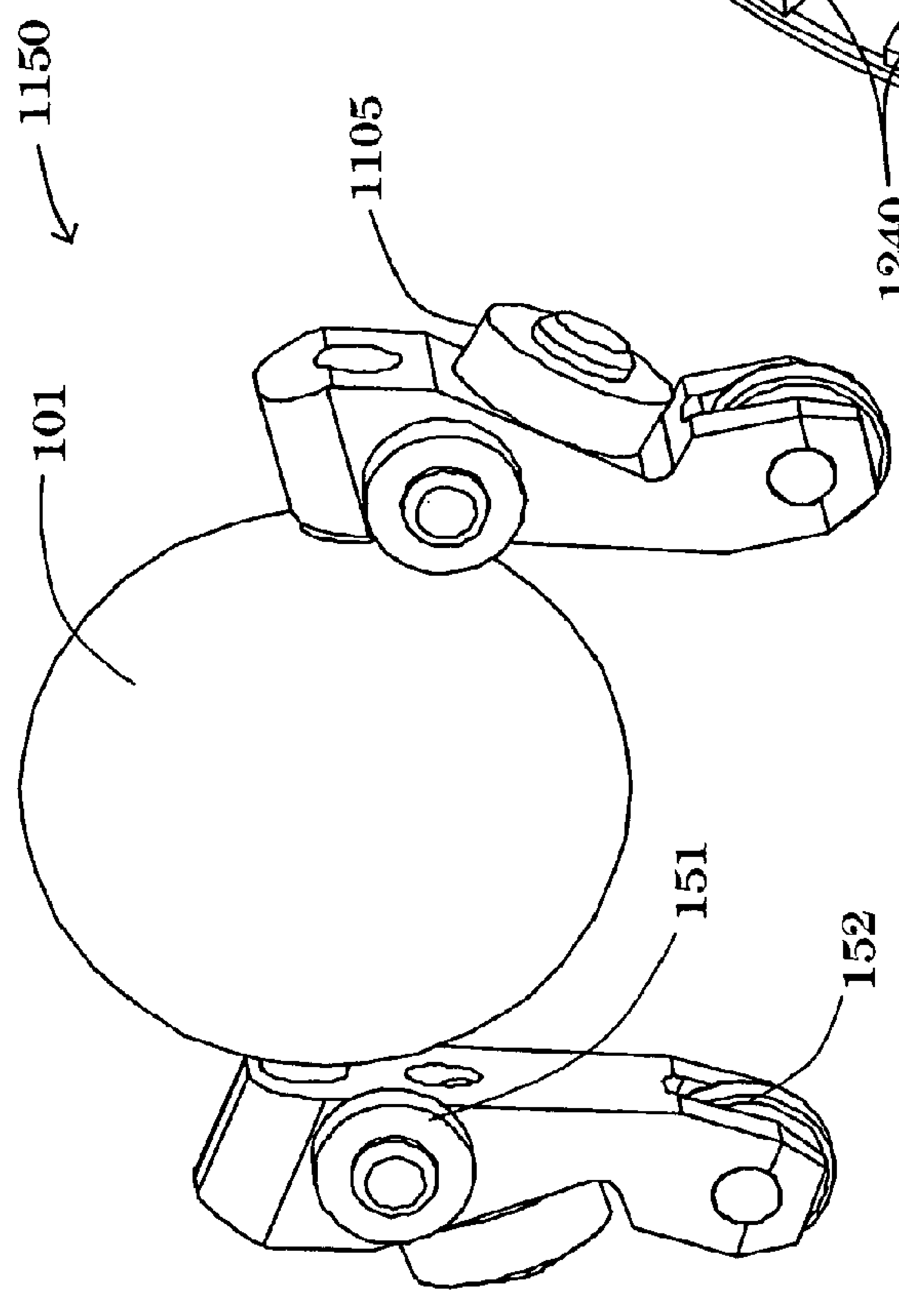

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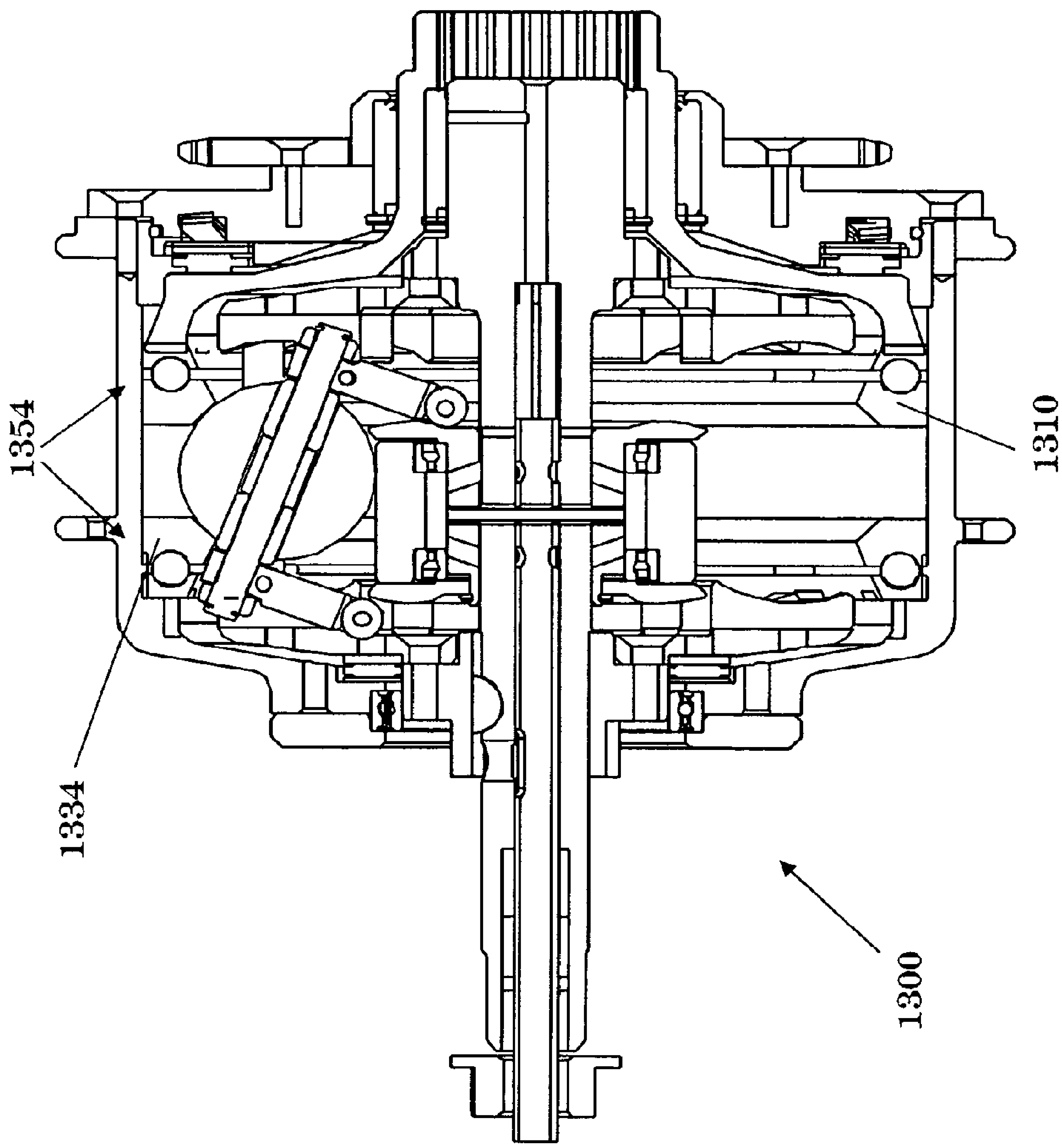

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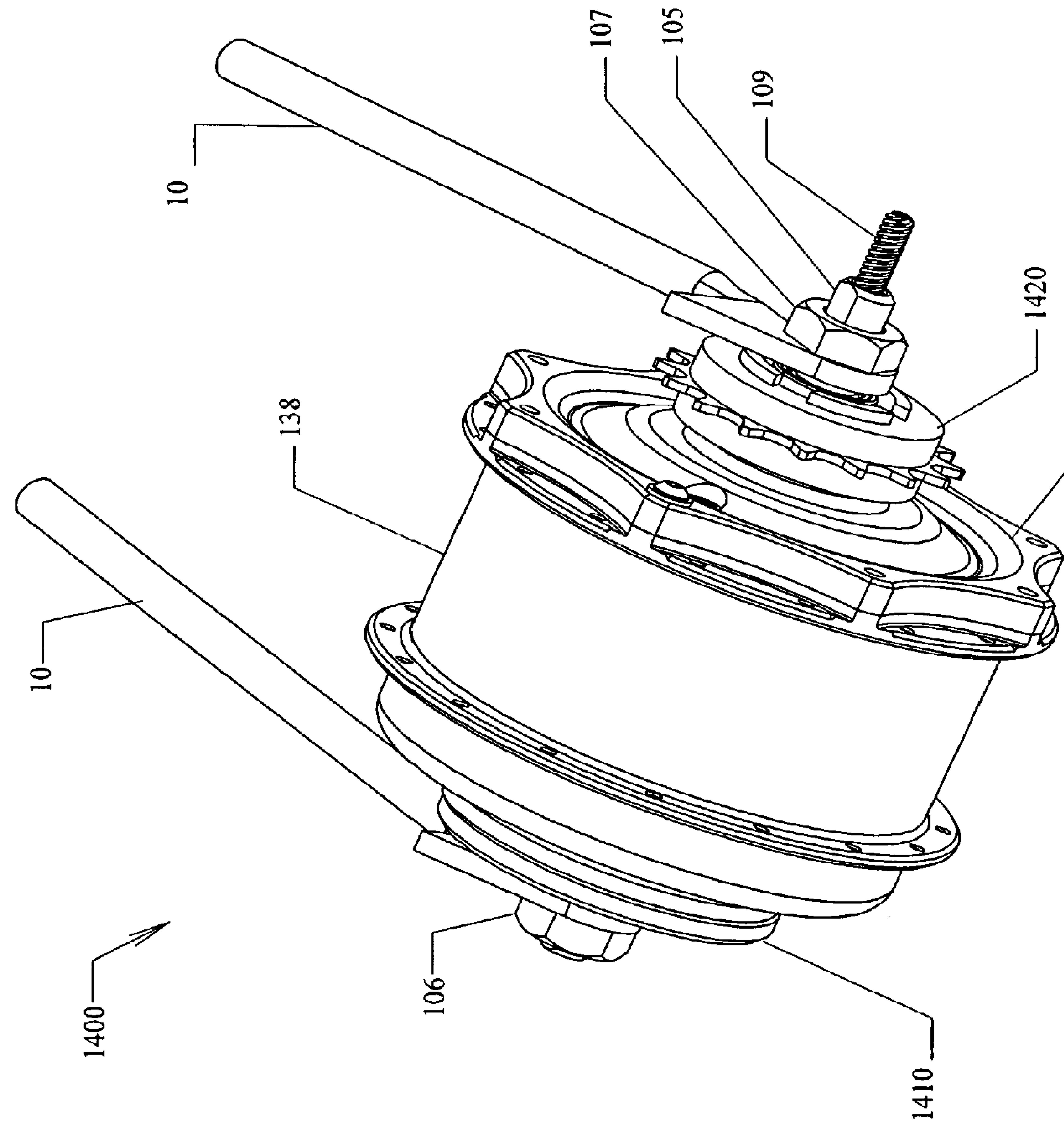

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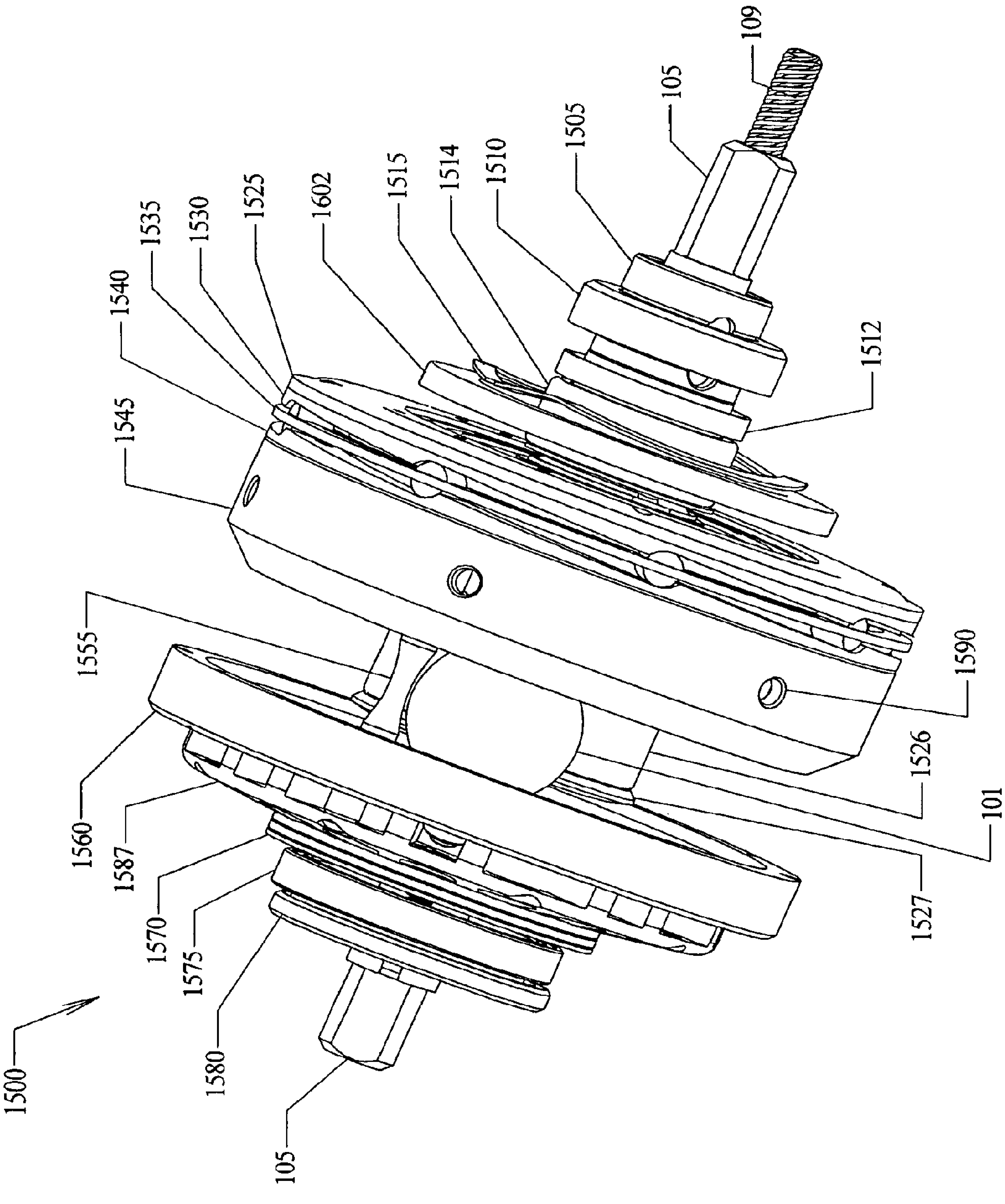
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FIGURE 15

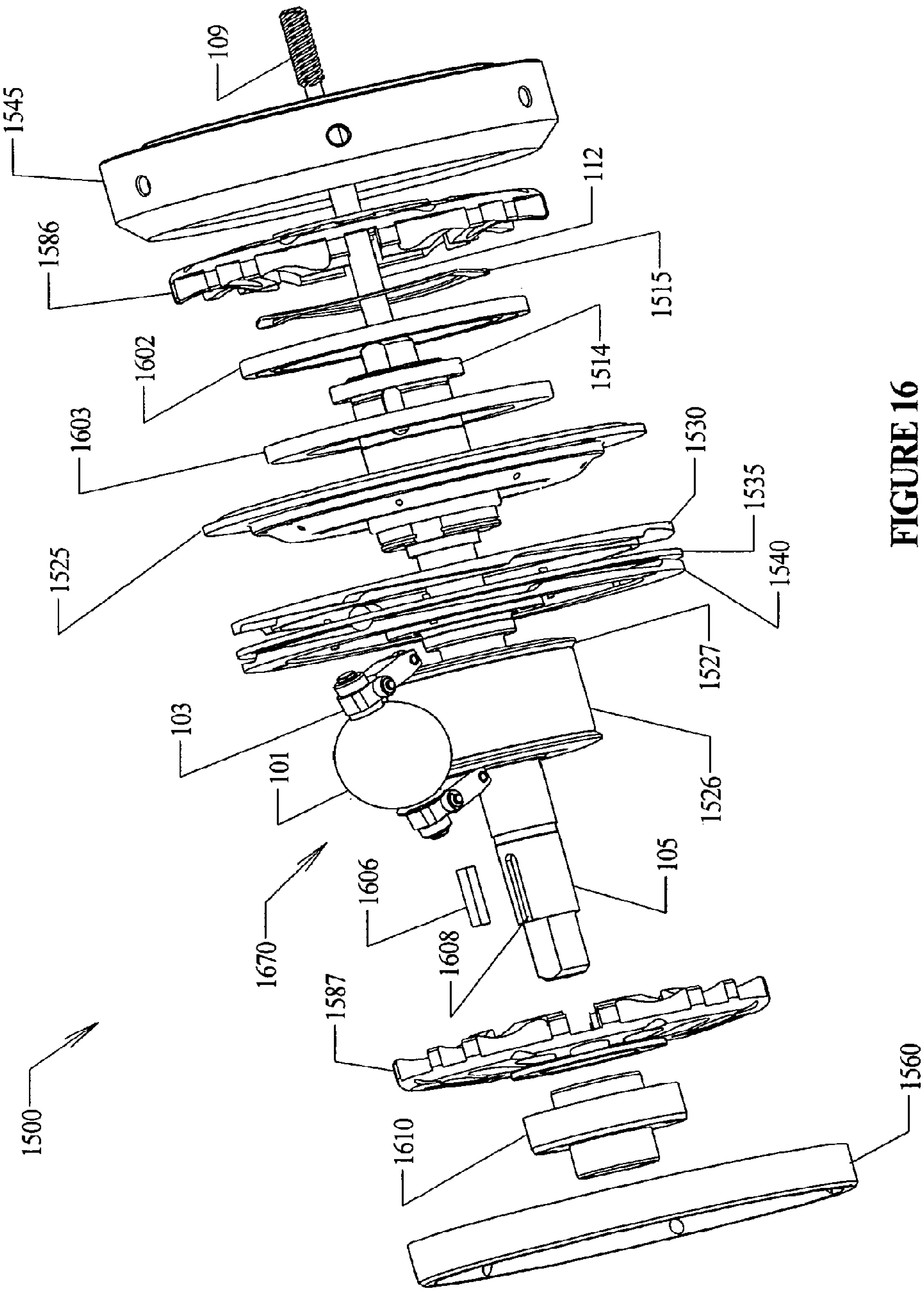
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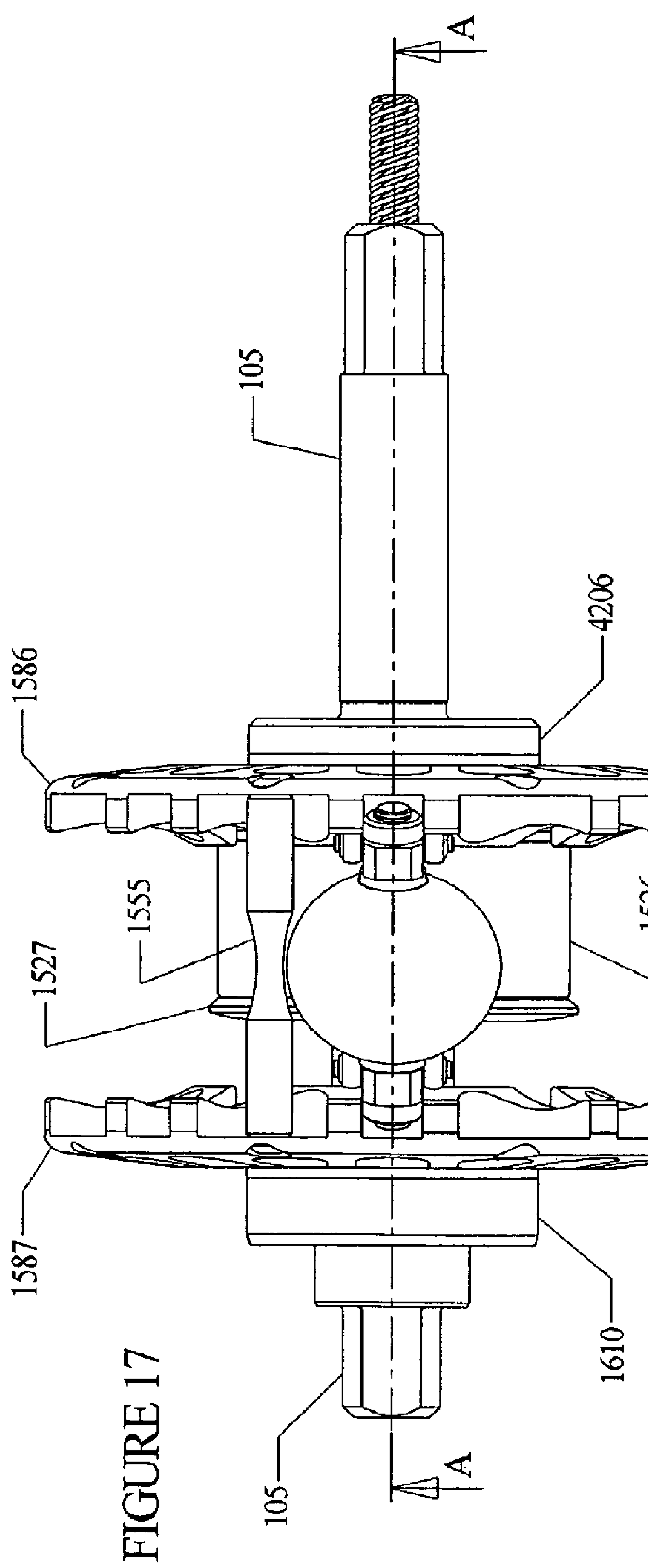
FIGURE 16



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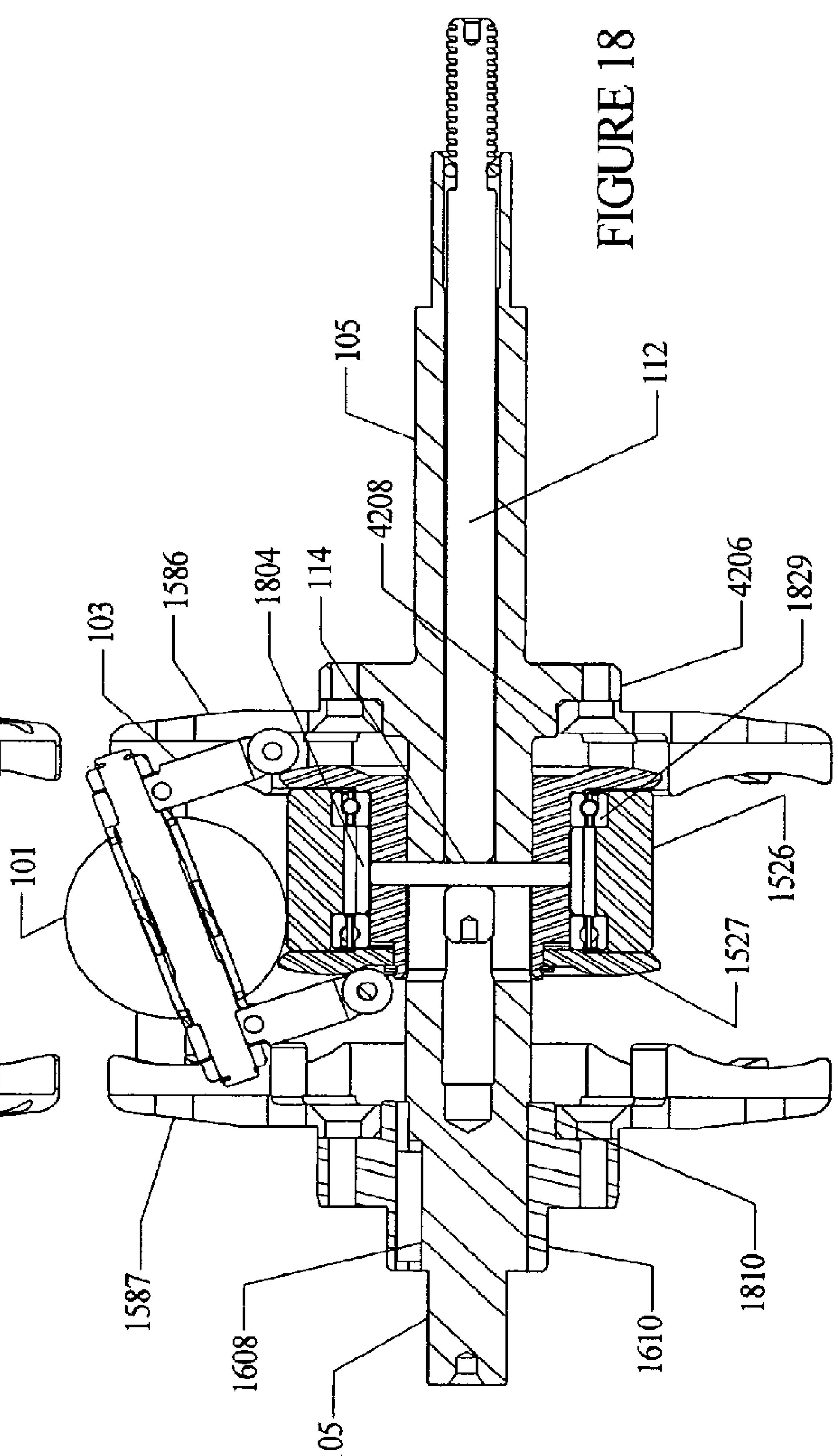

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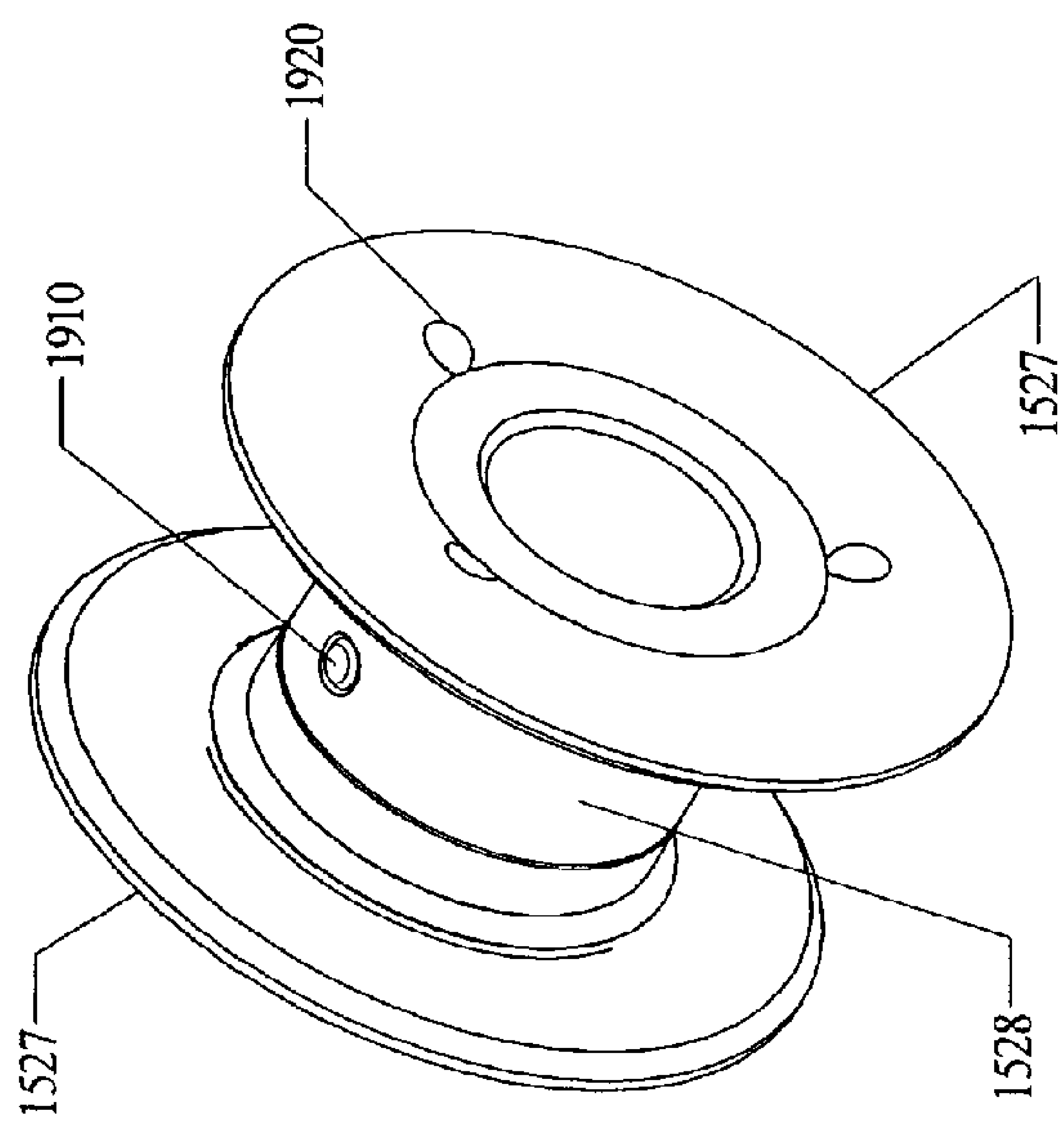

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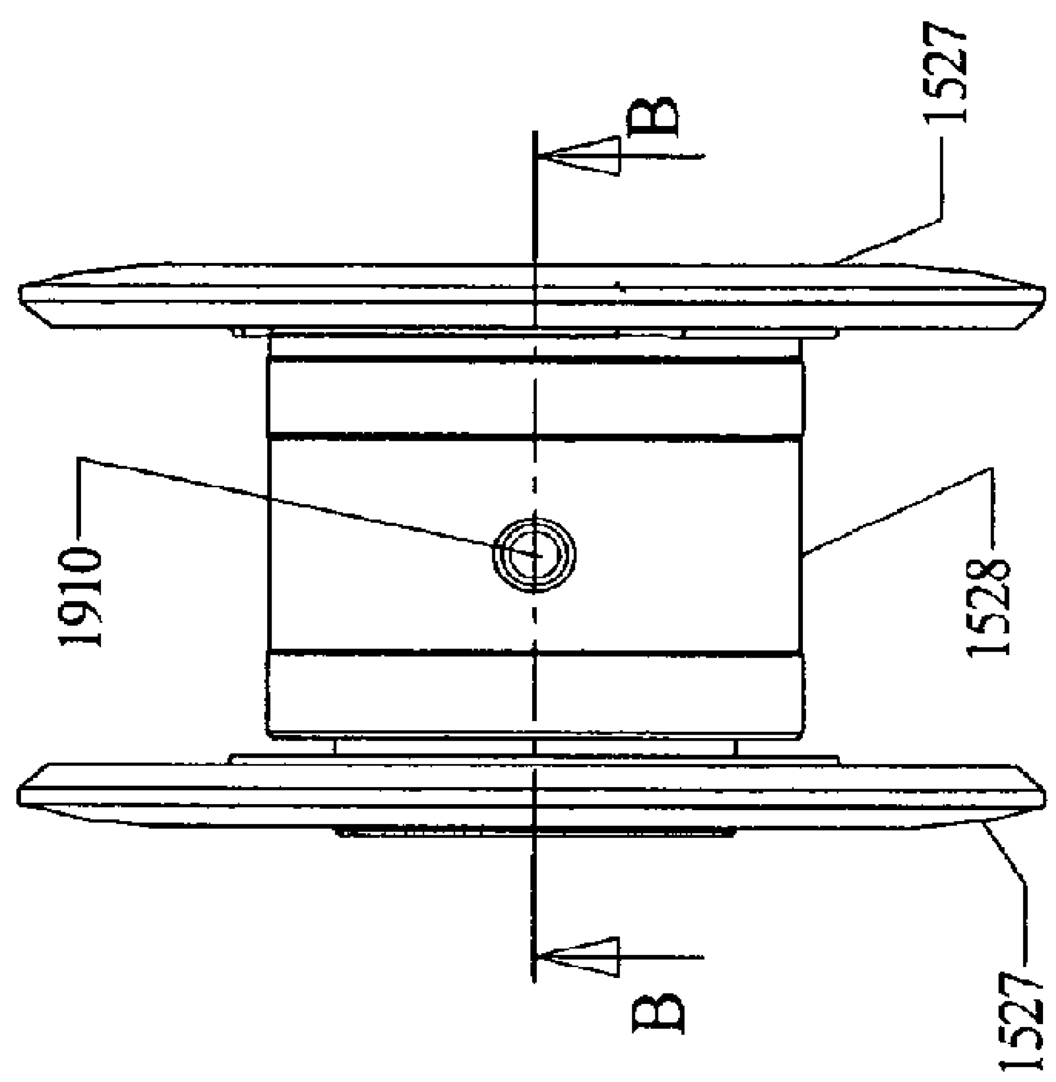

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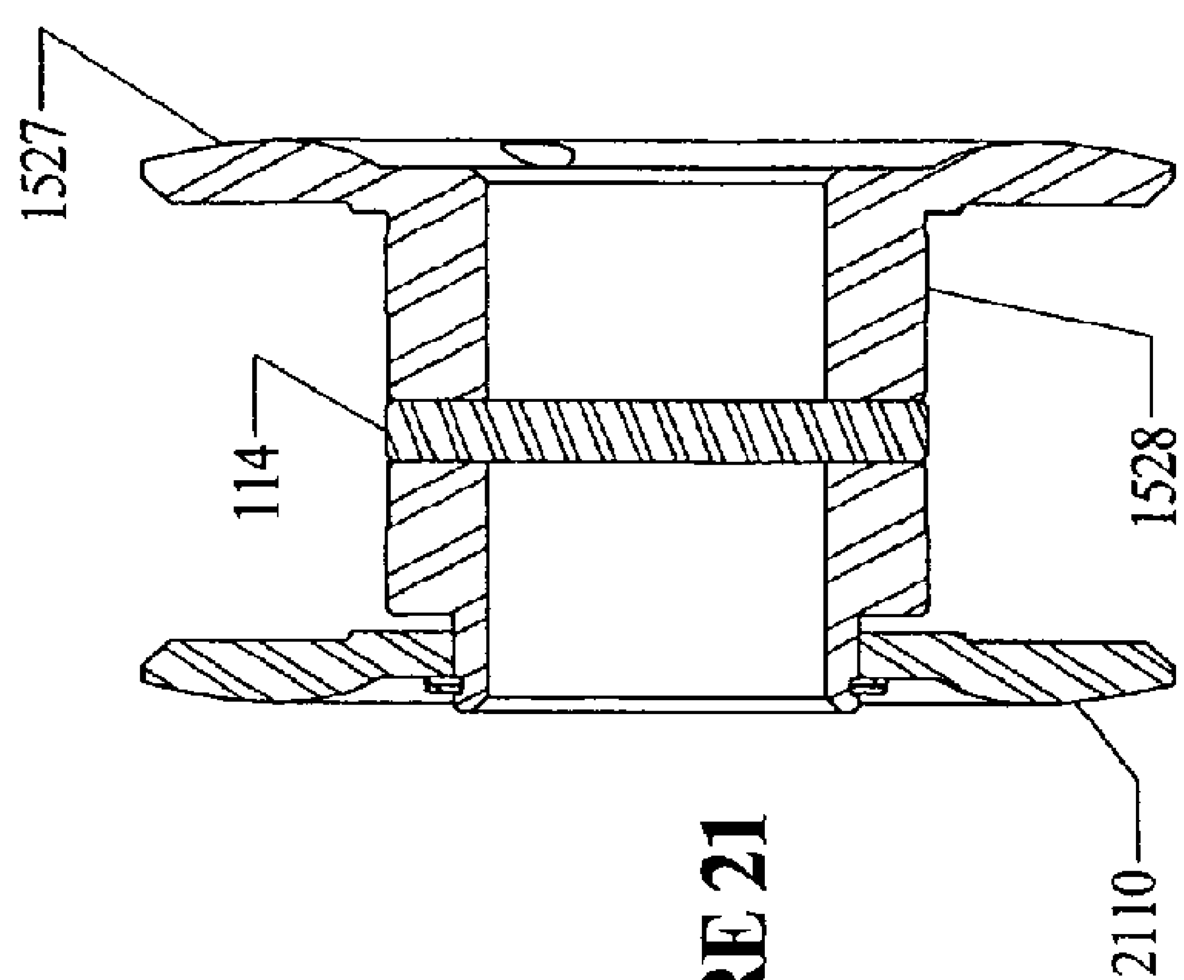

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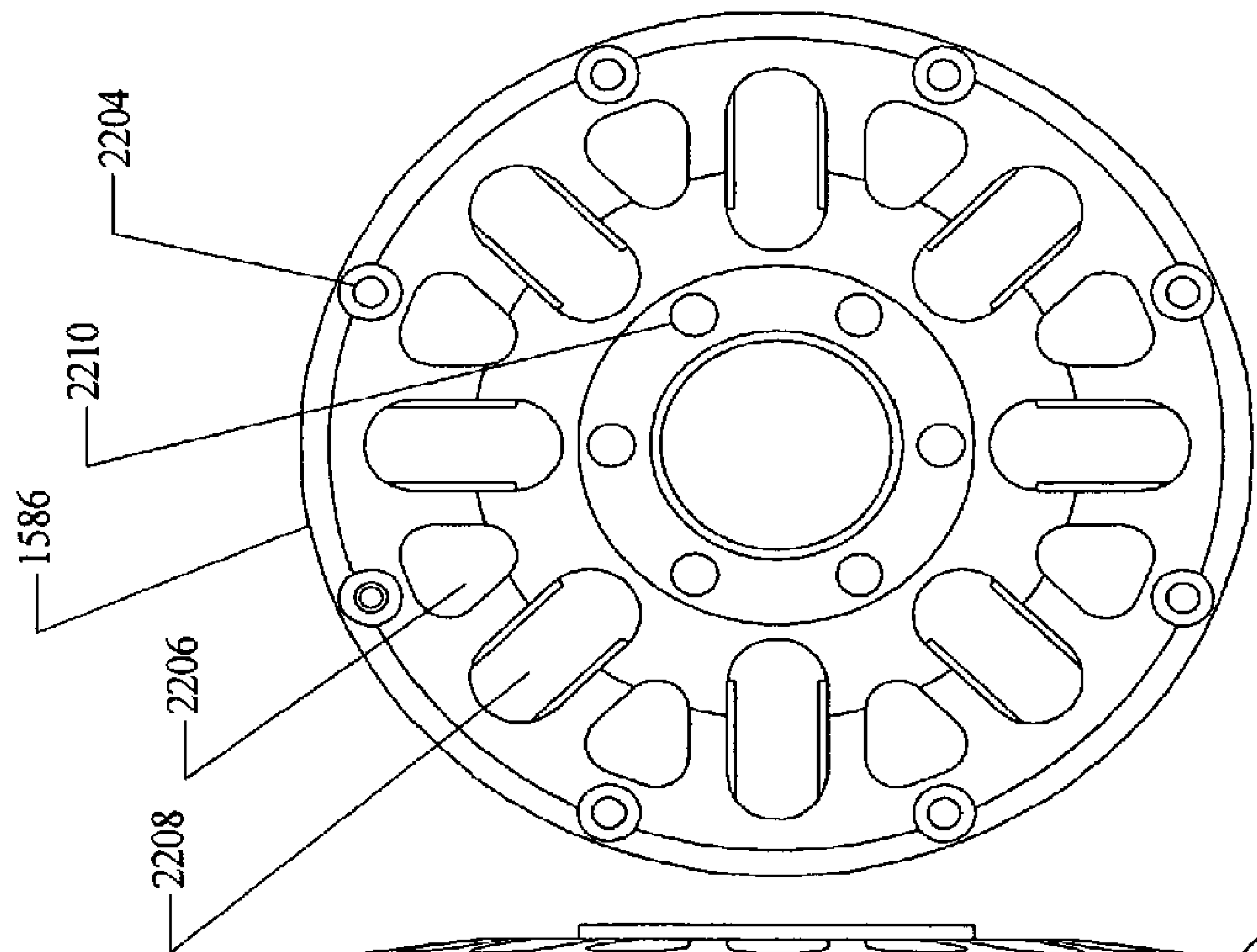

FIGURE 24

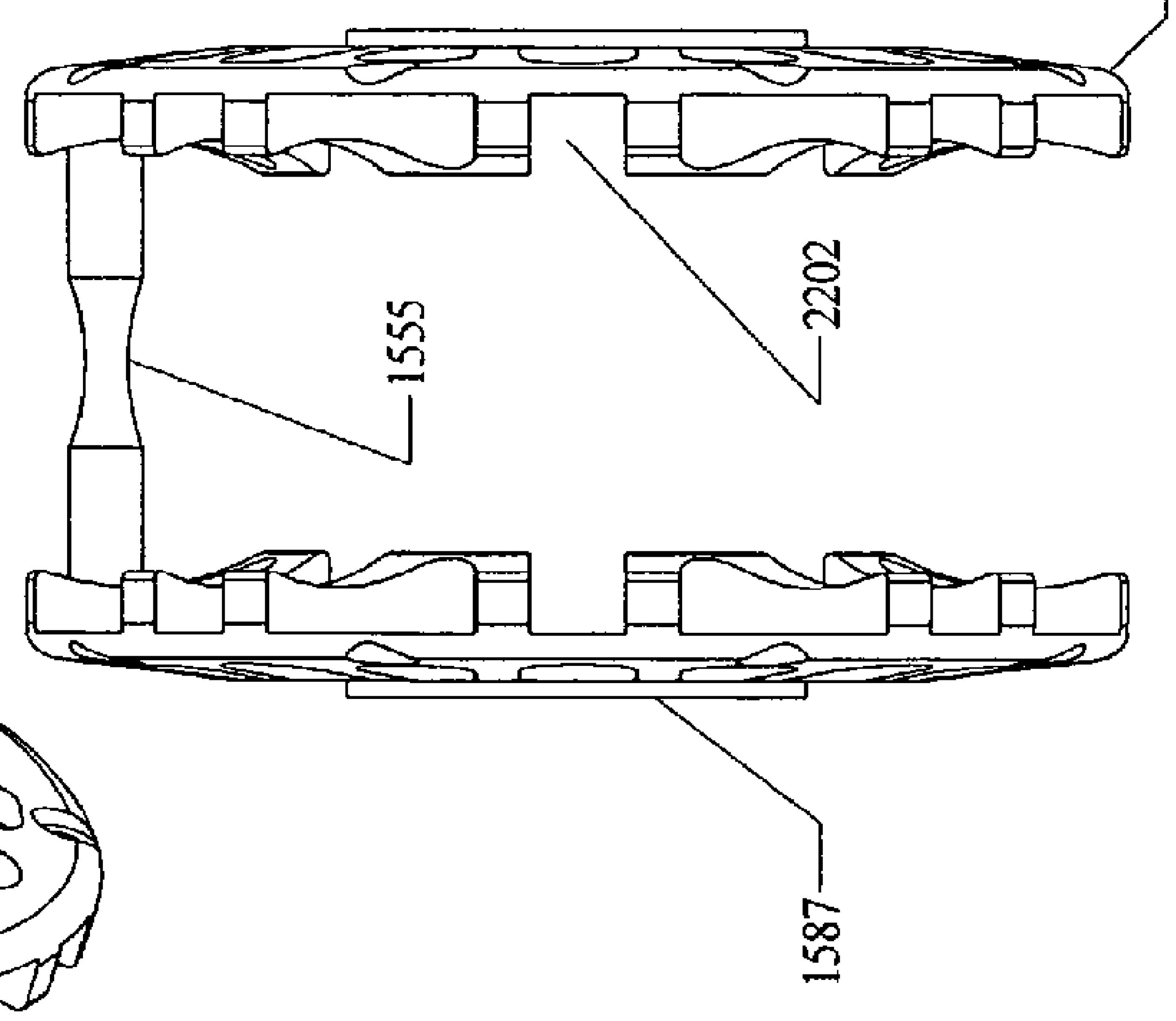

FIGURE 23

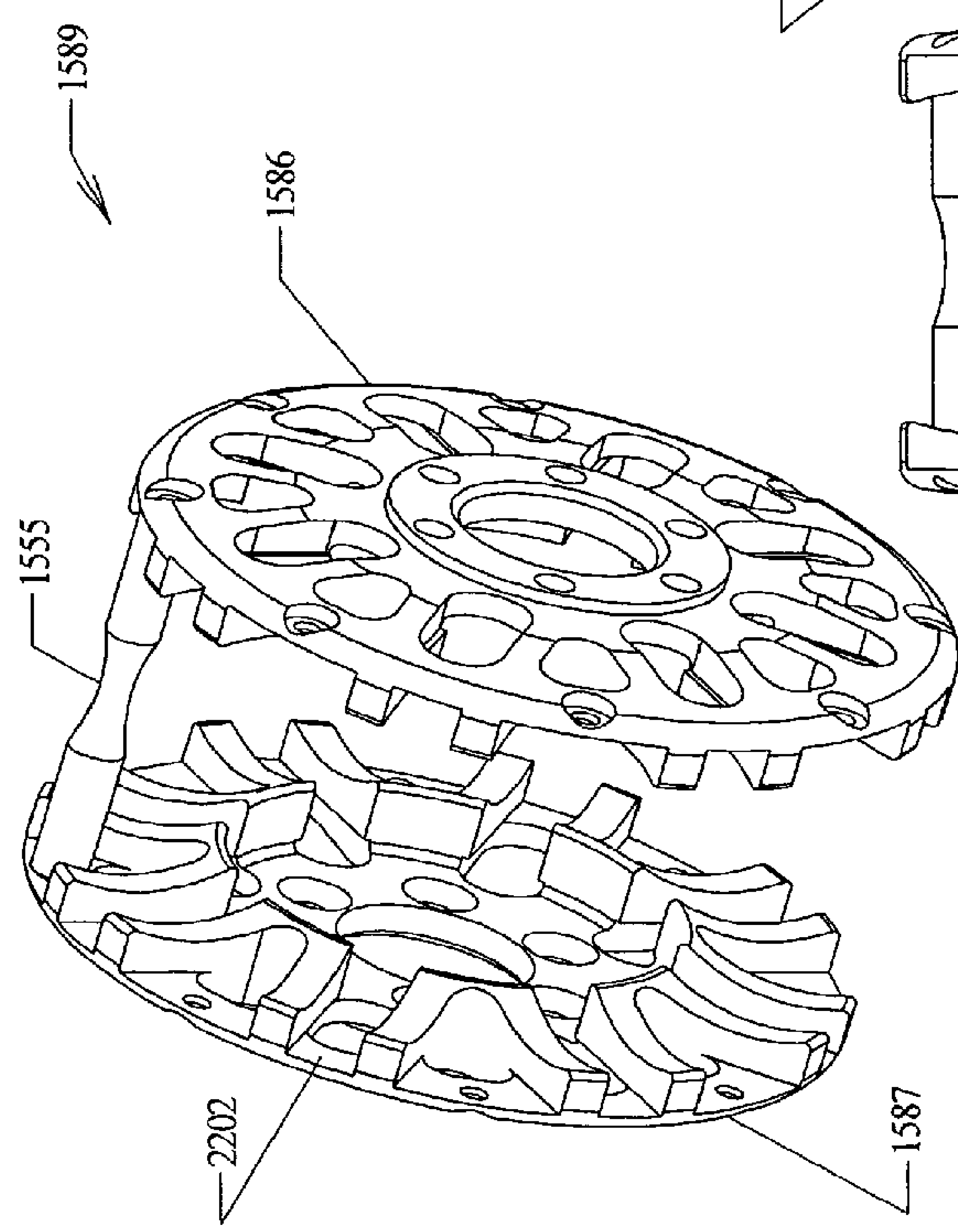

FIGURE 22

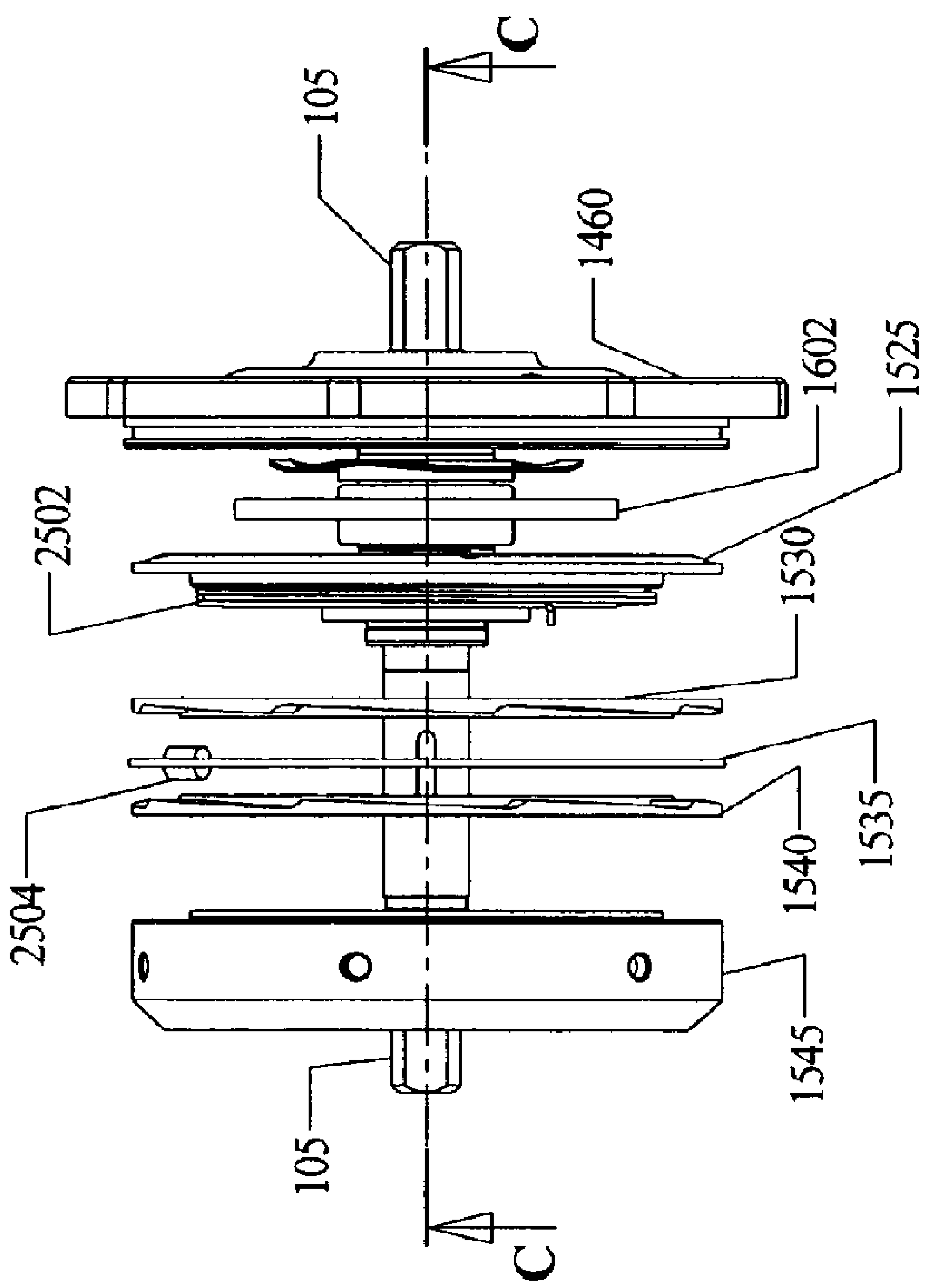

FIGURE 25

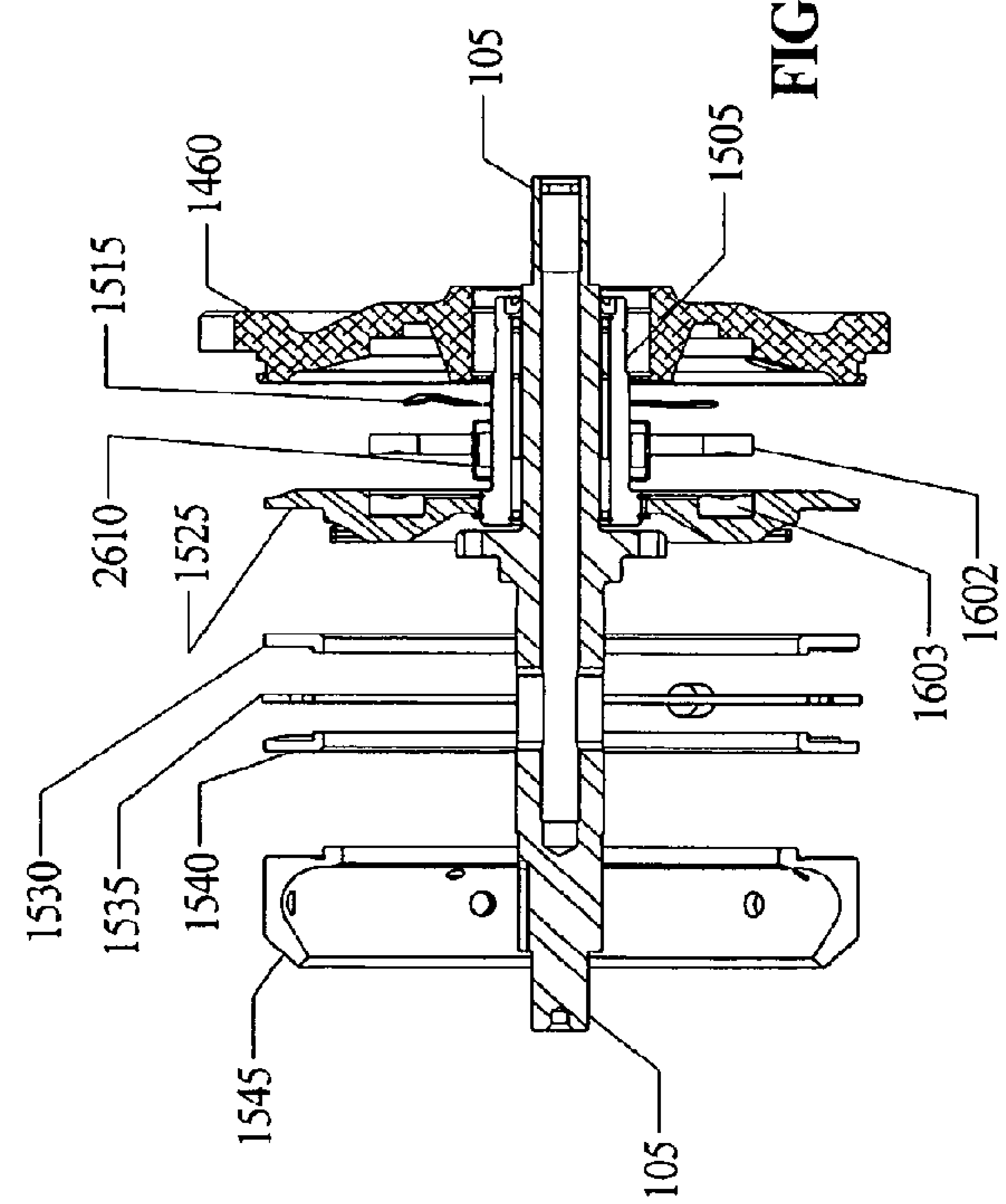

FIGURE 26

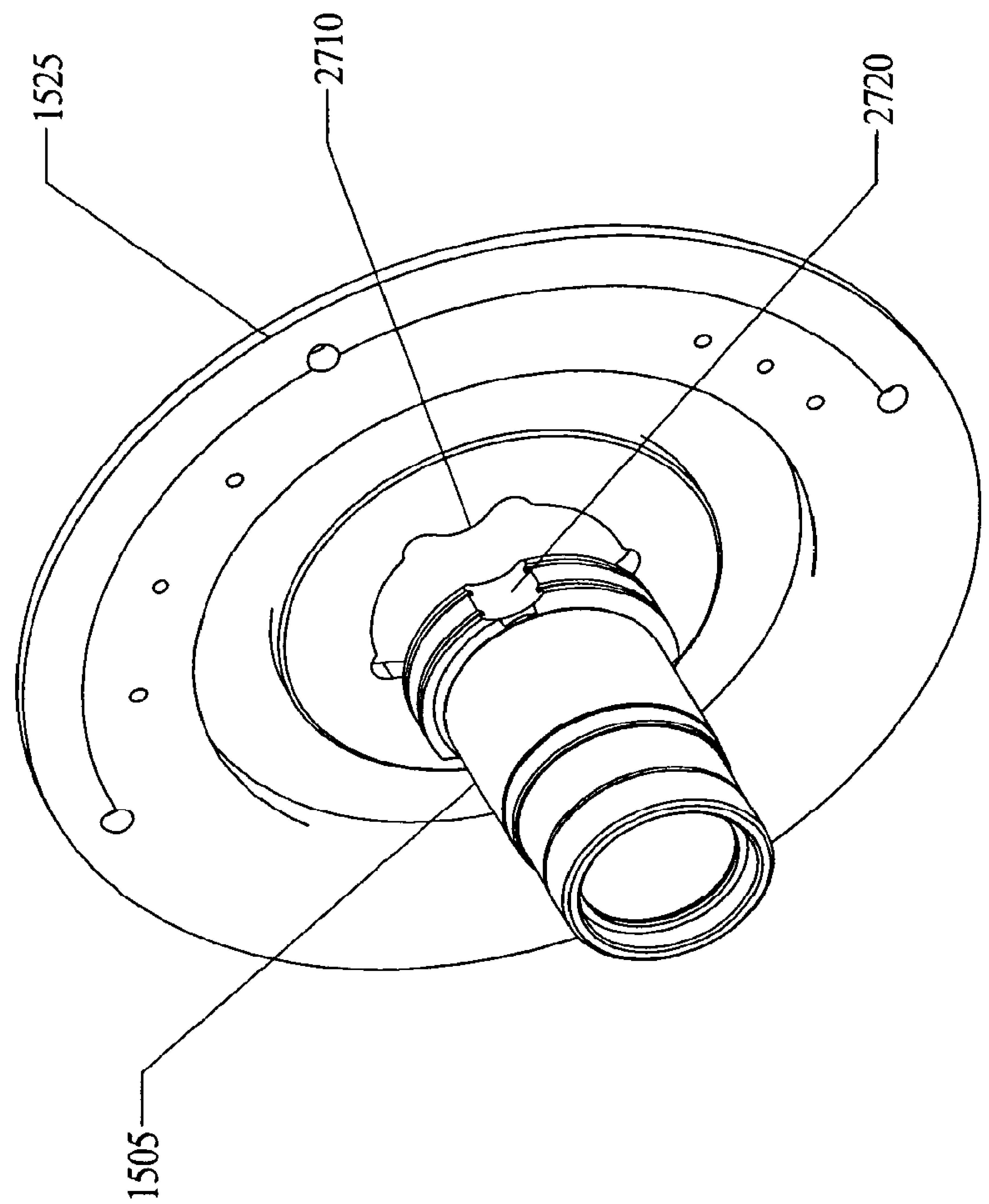

FIGURE 27

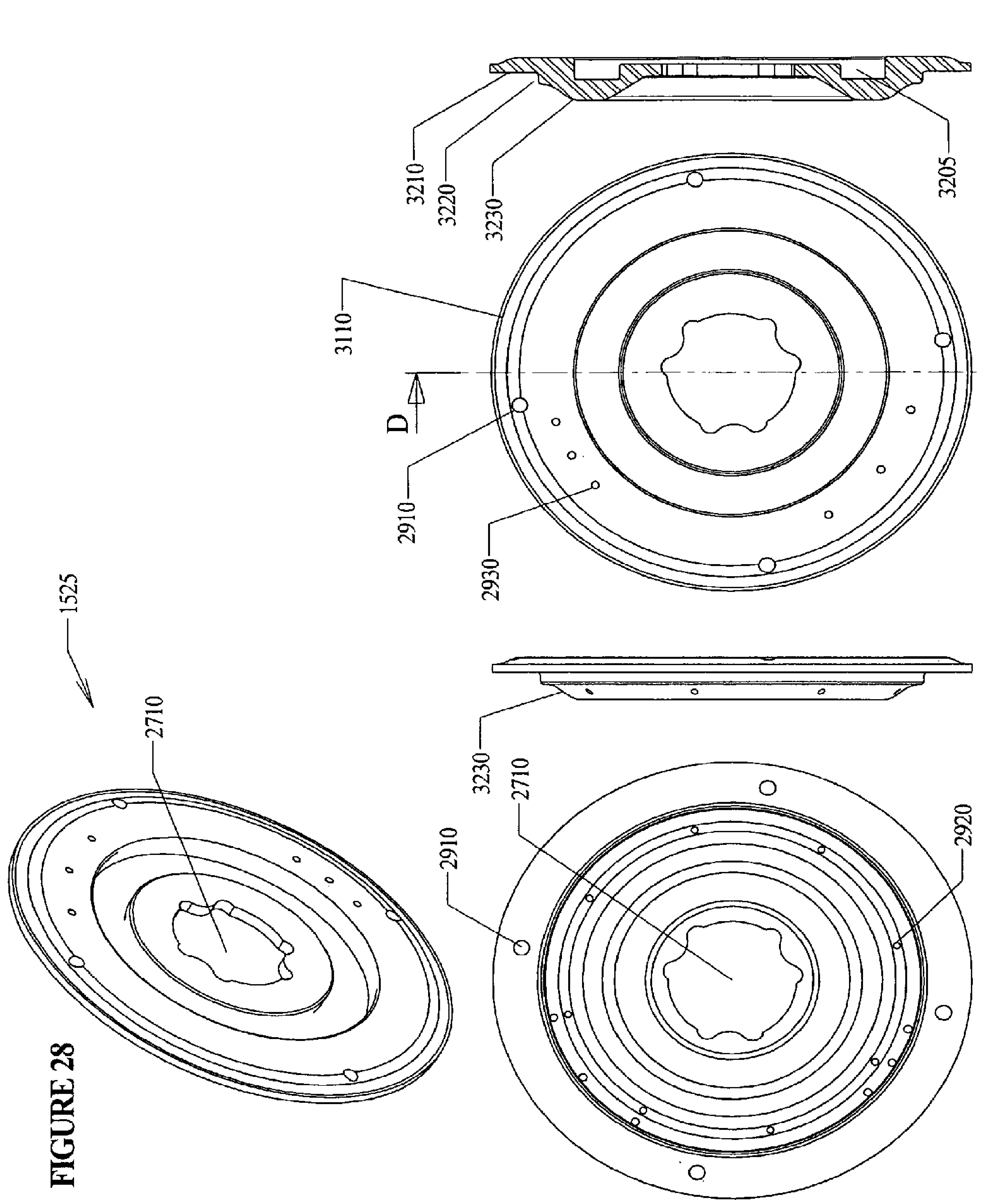

FIGURE 28

FIGURE 29

FIGURE 30

FIGURE 31

FIGURE 32

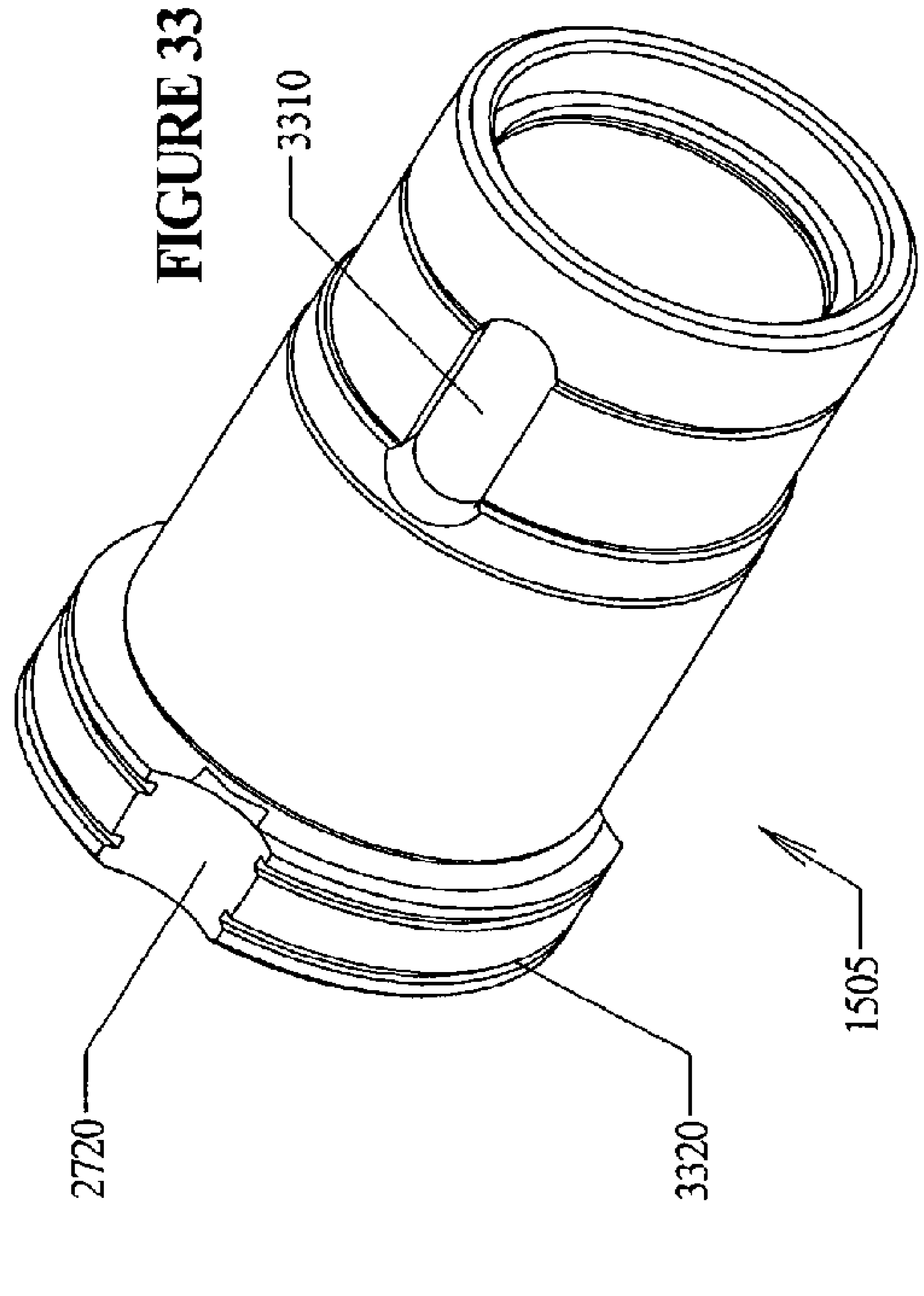

FIGURE 33

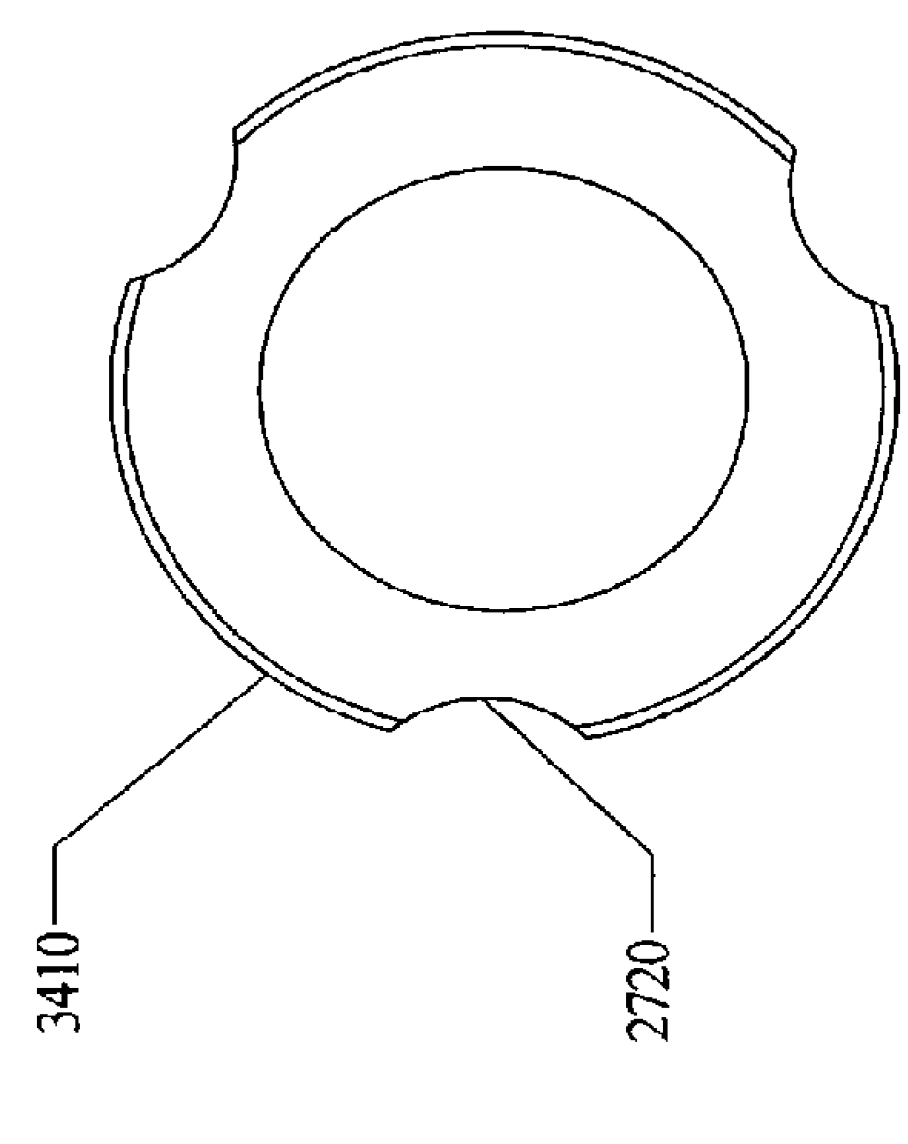

FIGURE 34

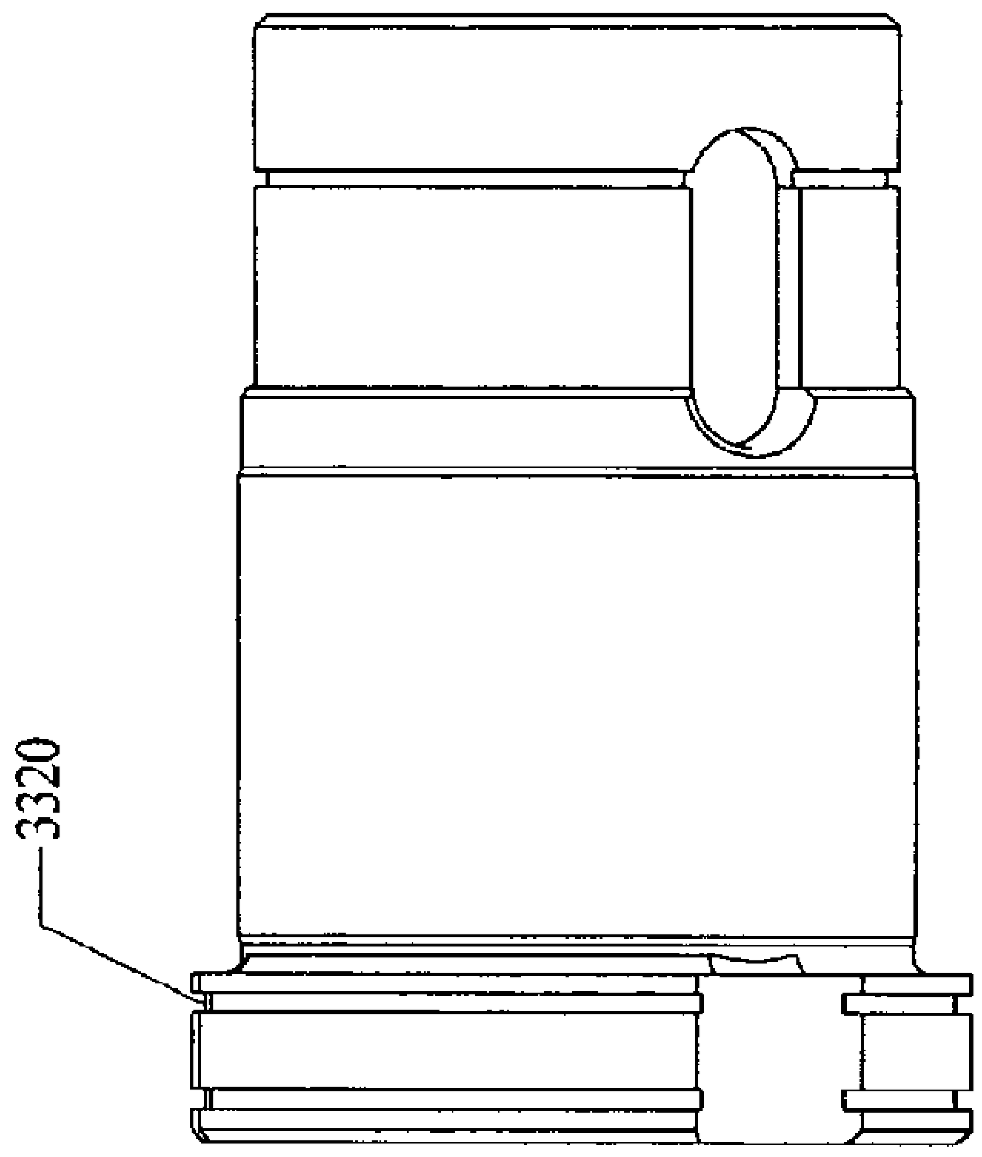

FIGURE 35

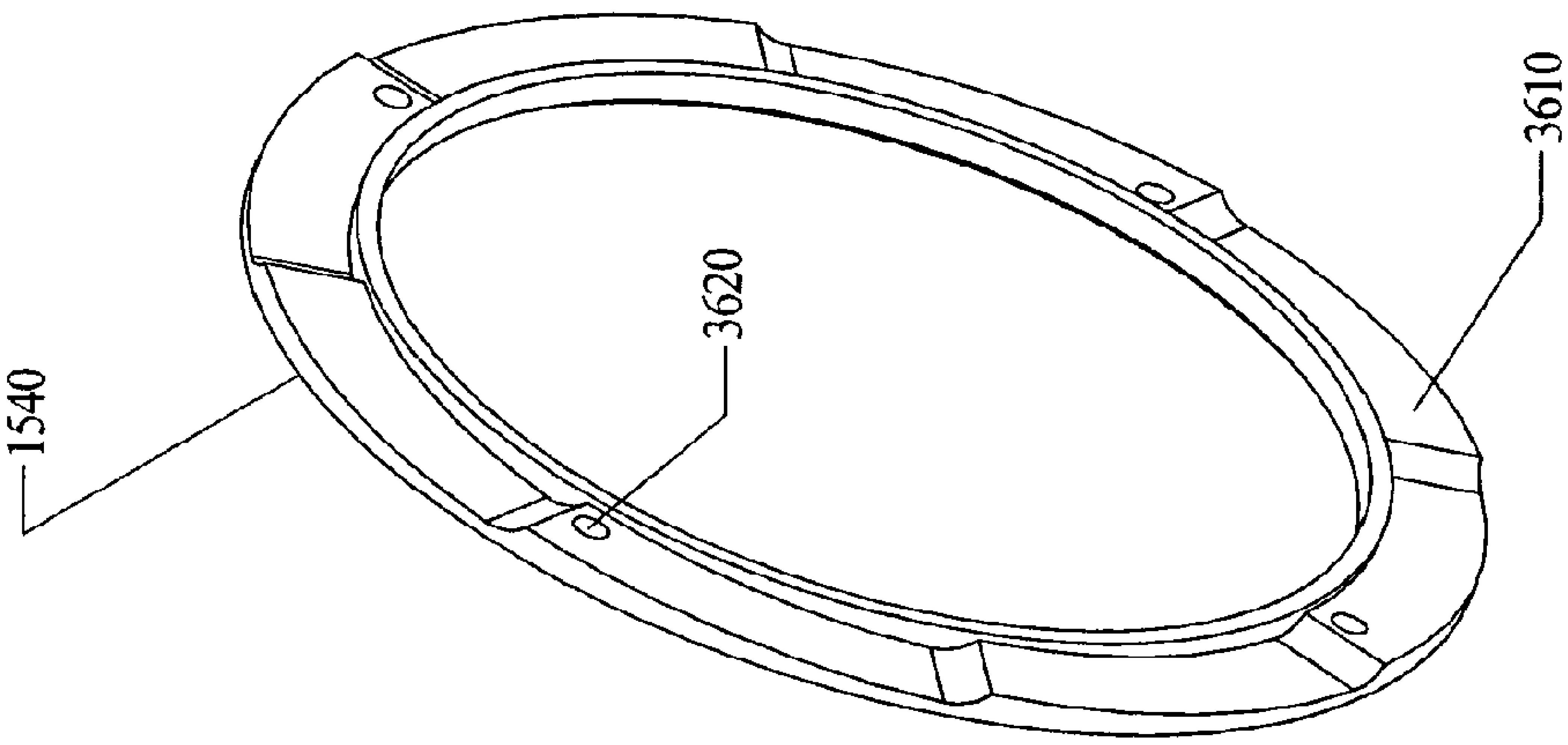

FIGURE 36

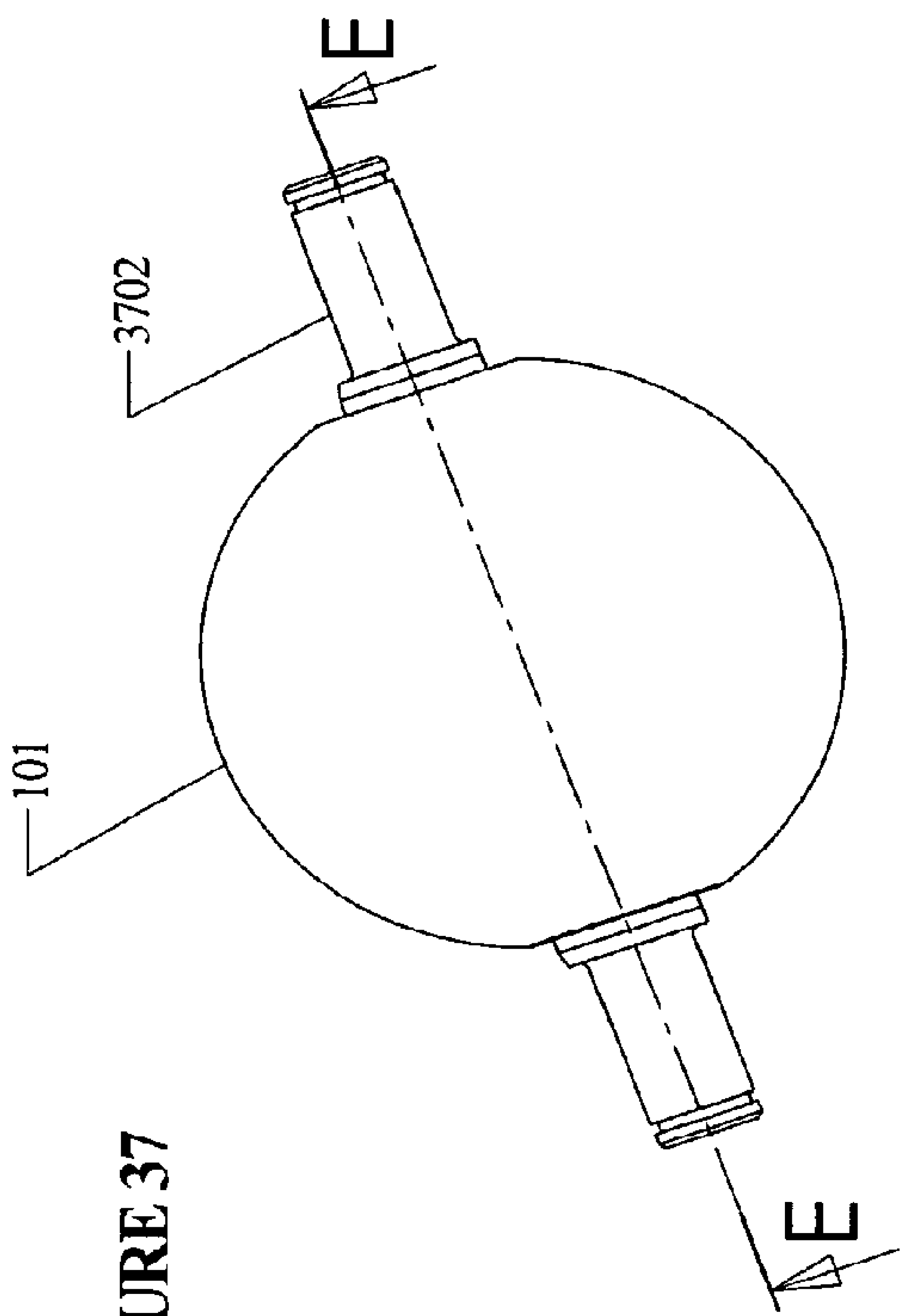

FIGURE 37

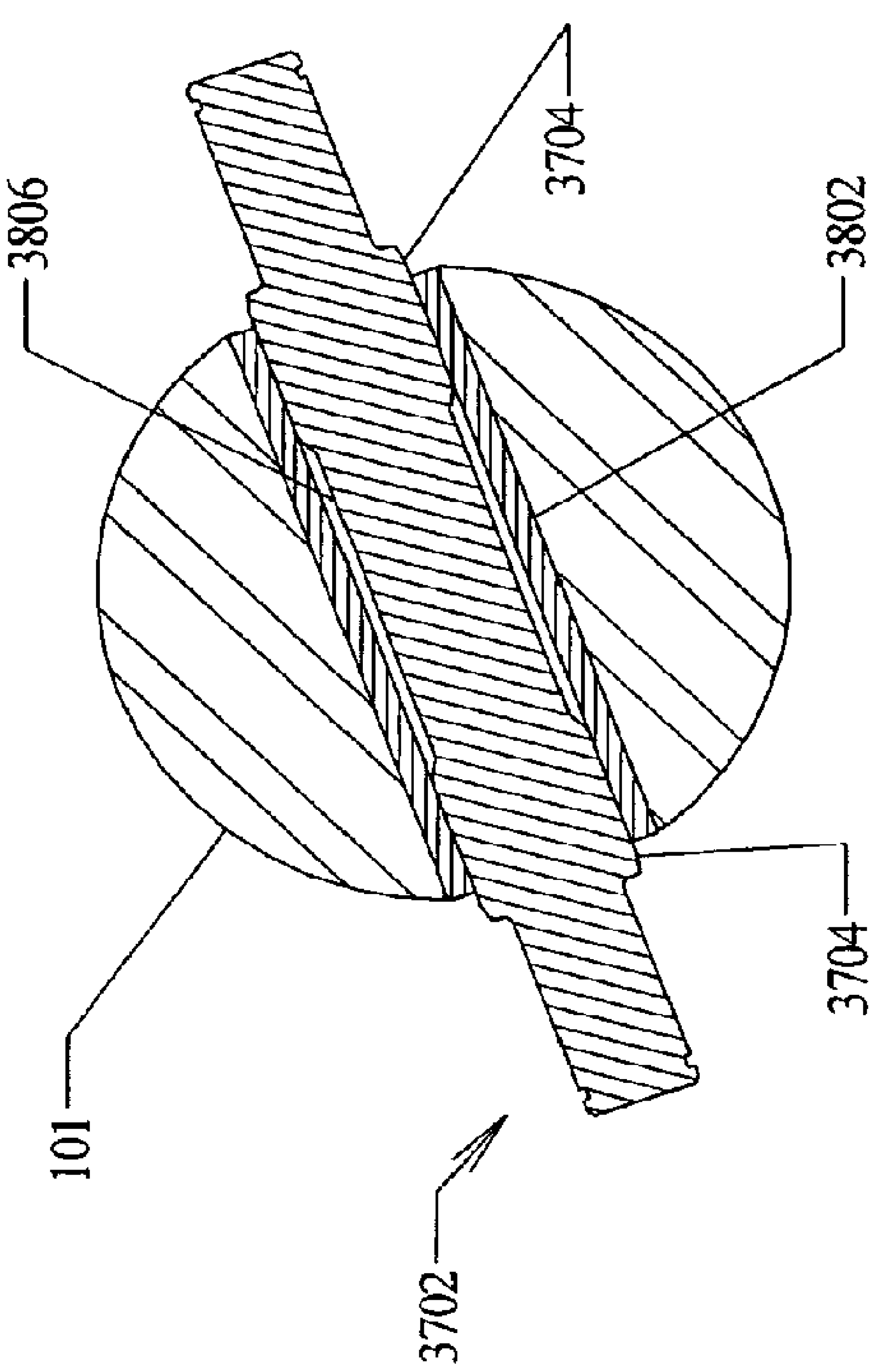

FIGURE 38

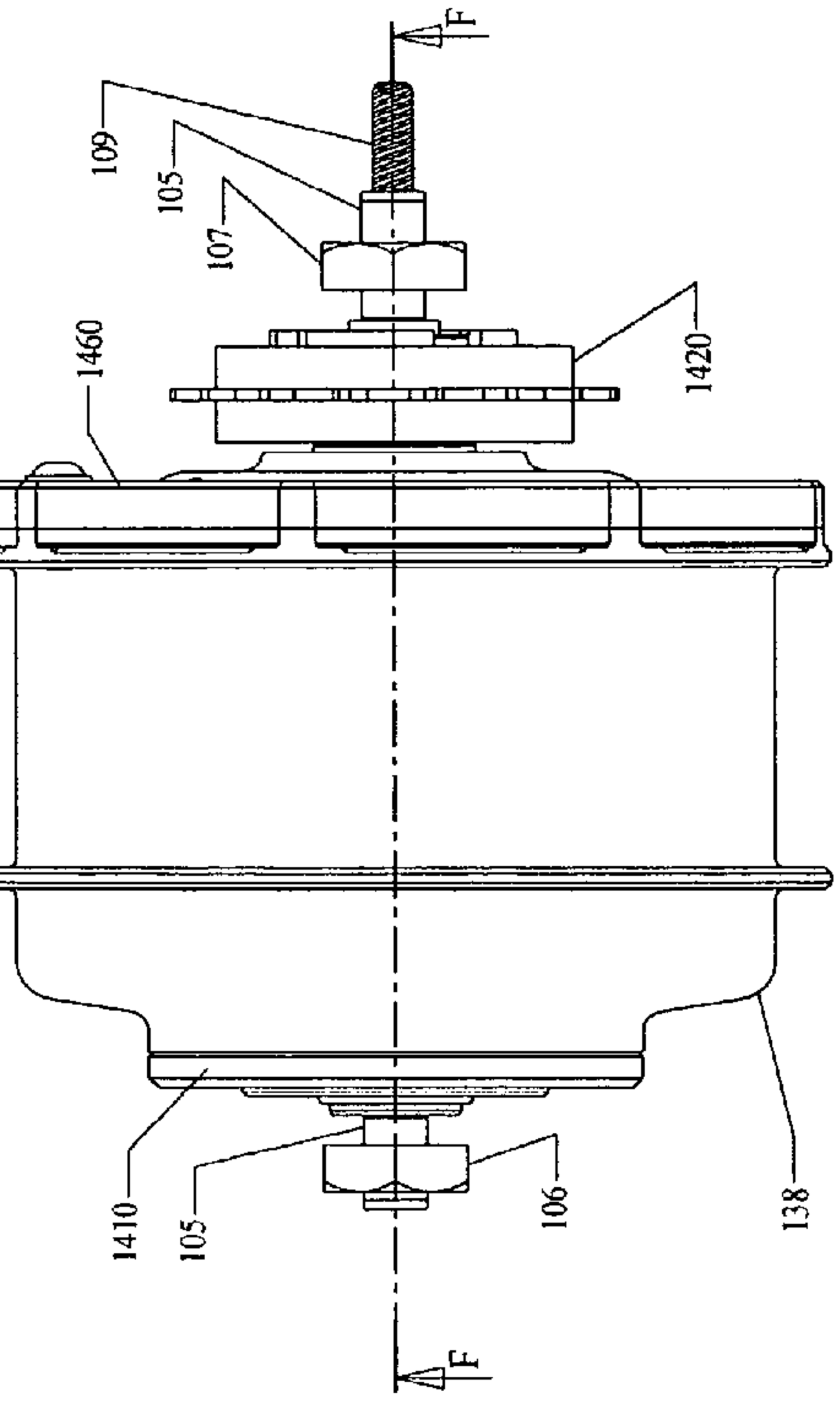

FIGURE 39

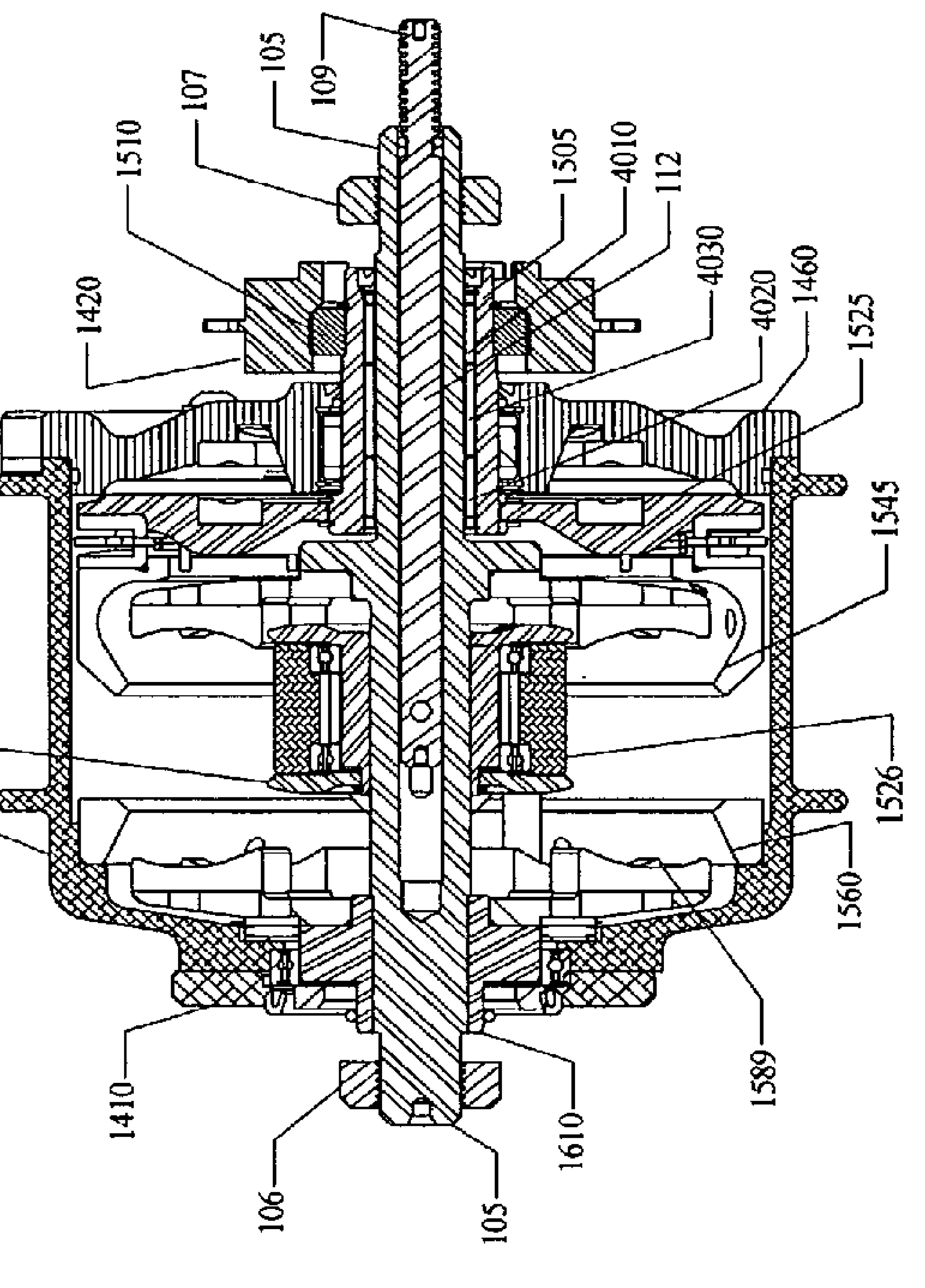

FIGURE 40

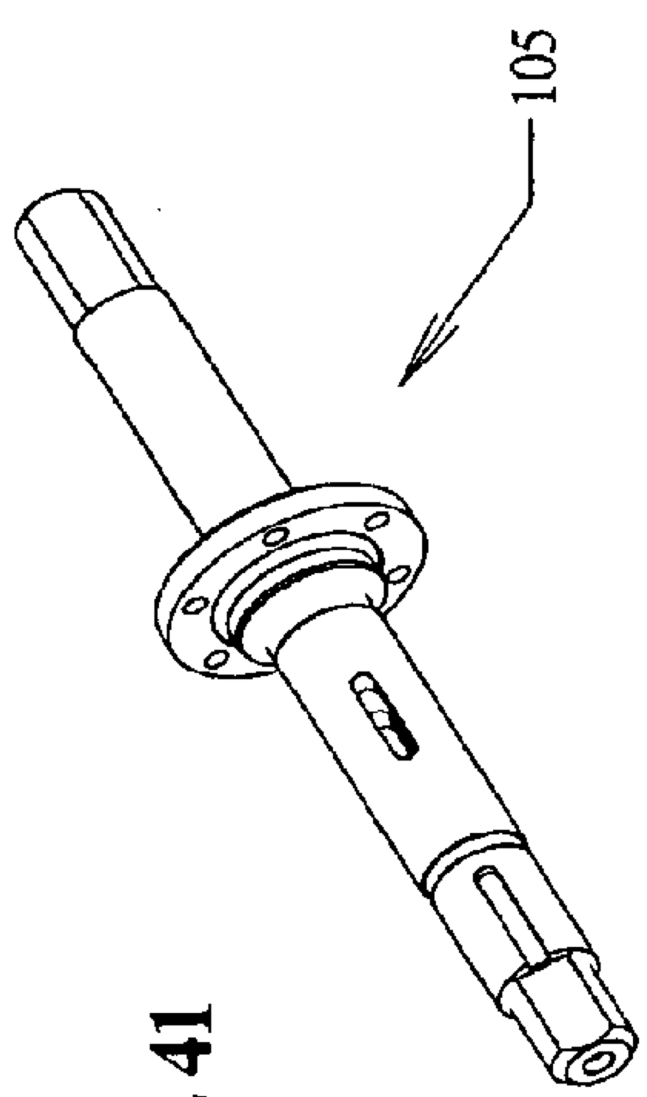

FIGURE 41

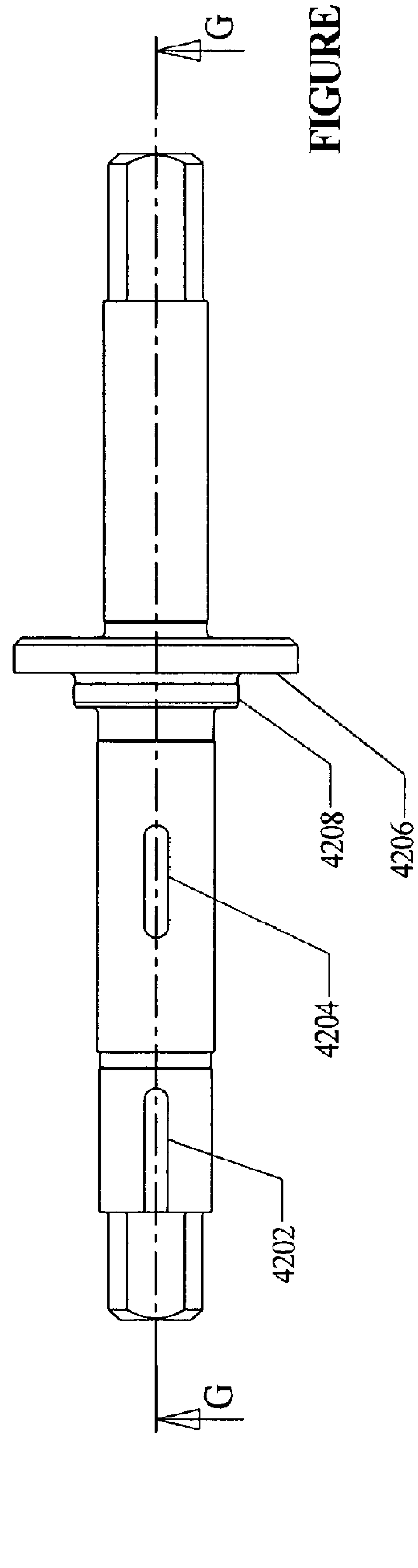

FIGURE 42

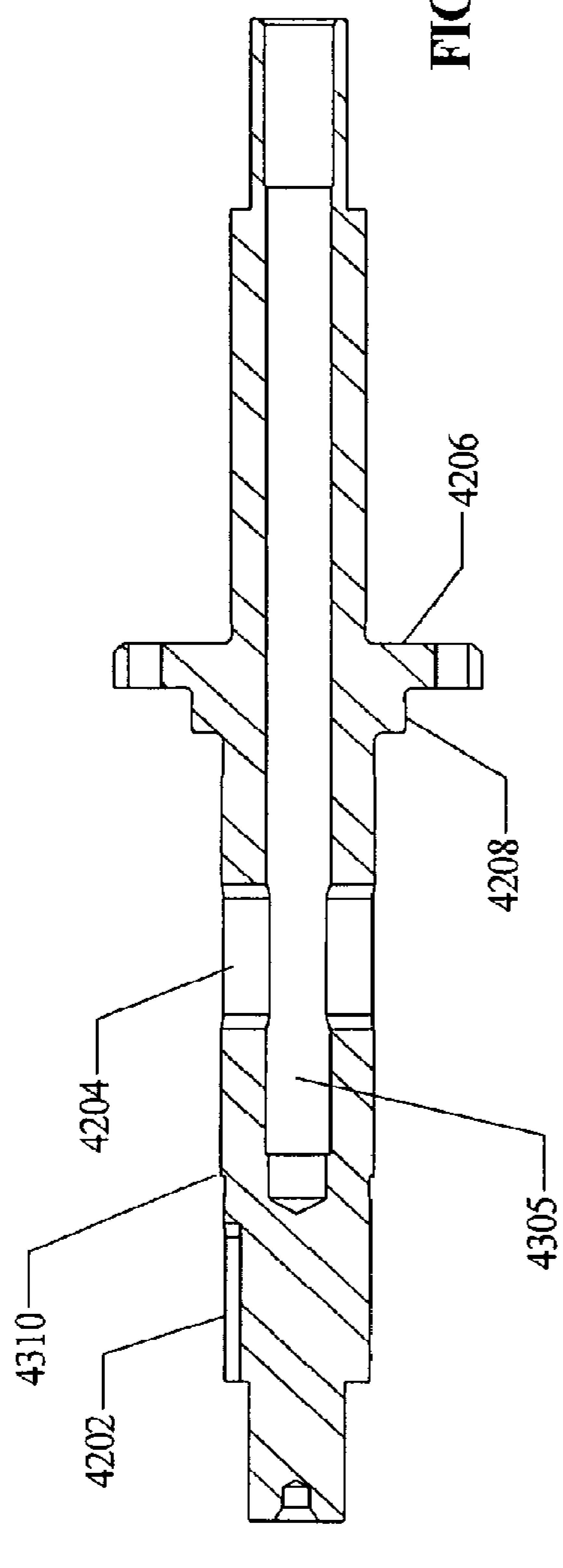

FIGURE 43

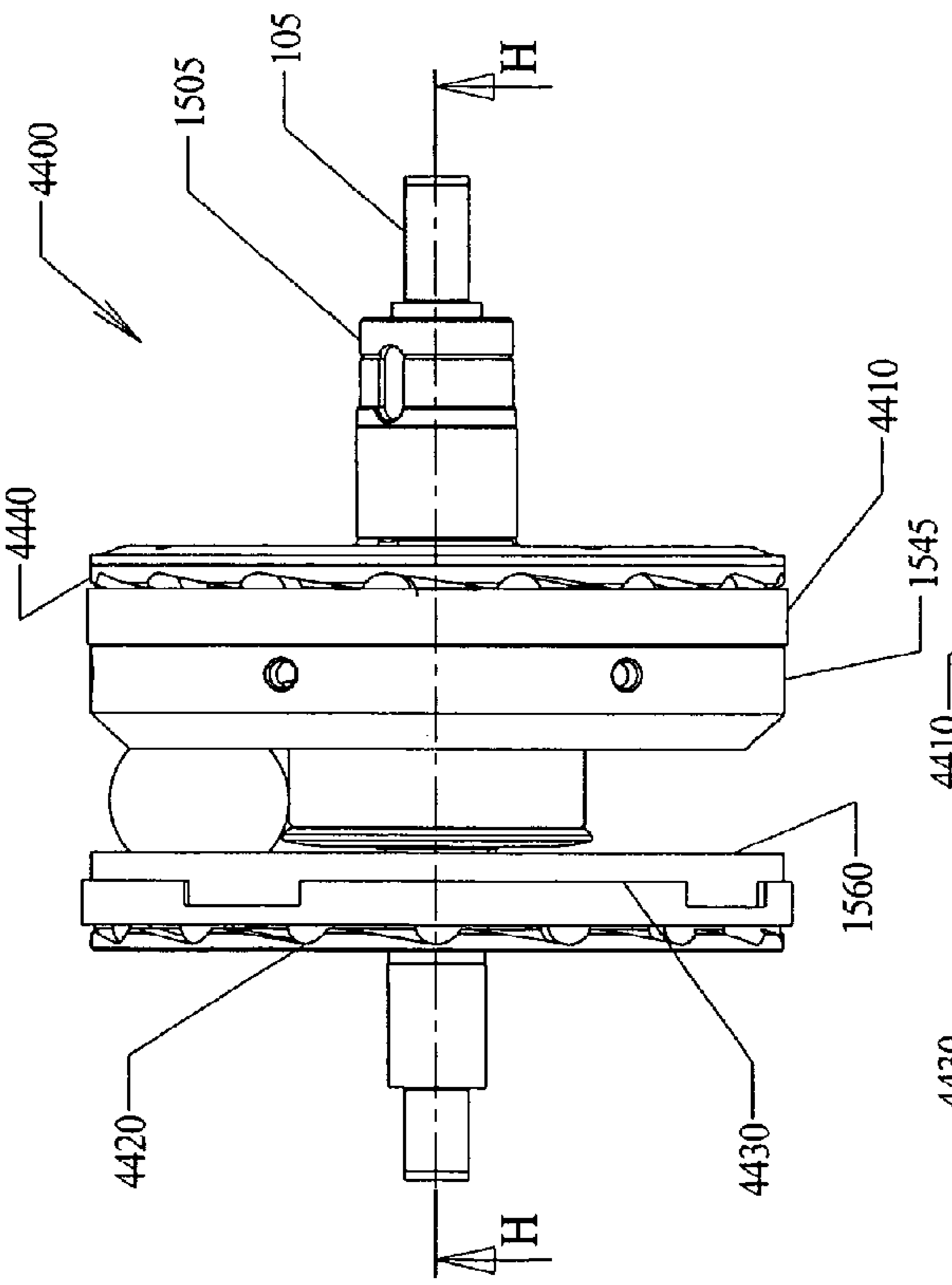

FIGURE 44

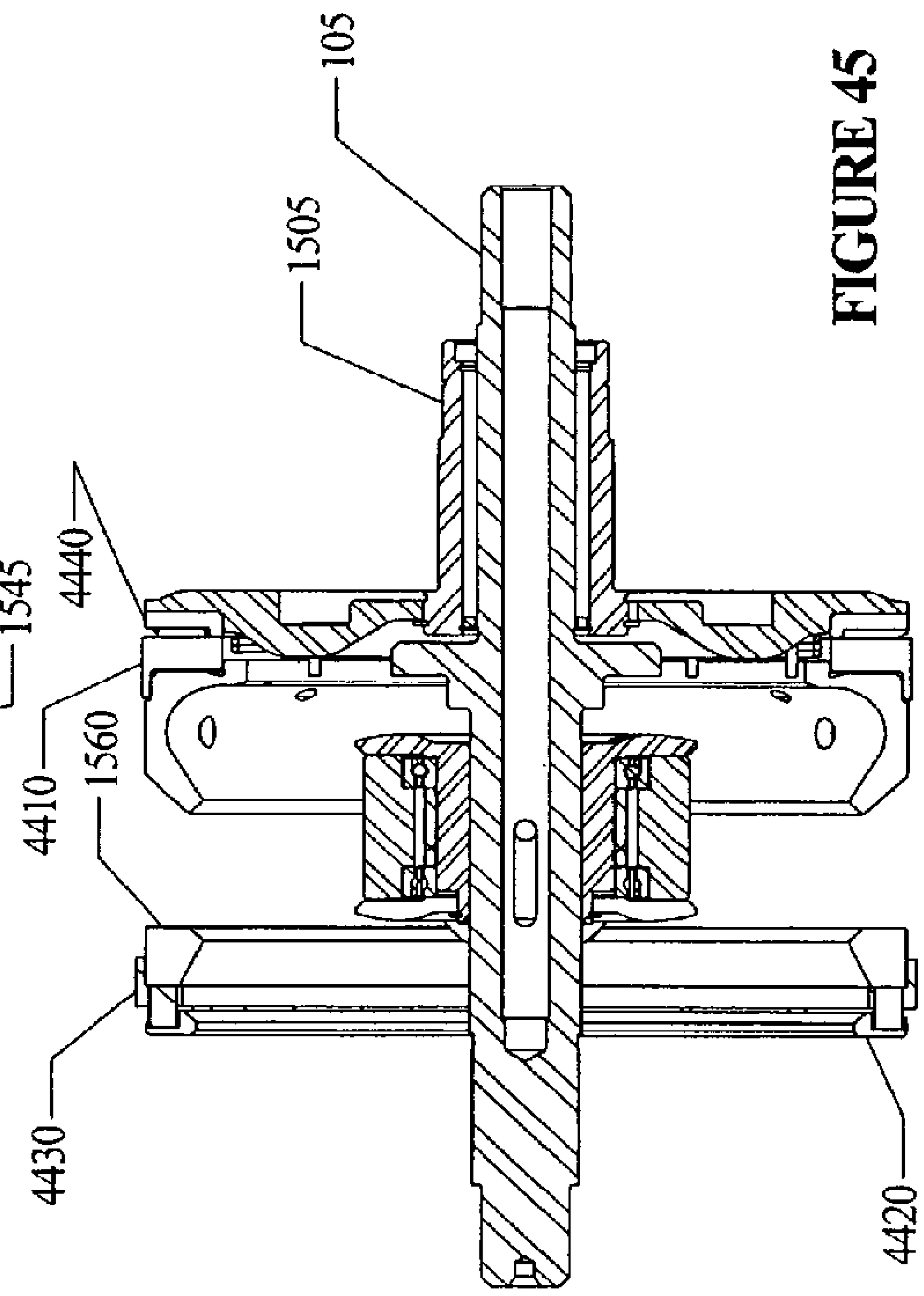

FIGURE 45

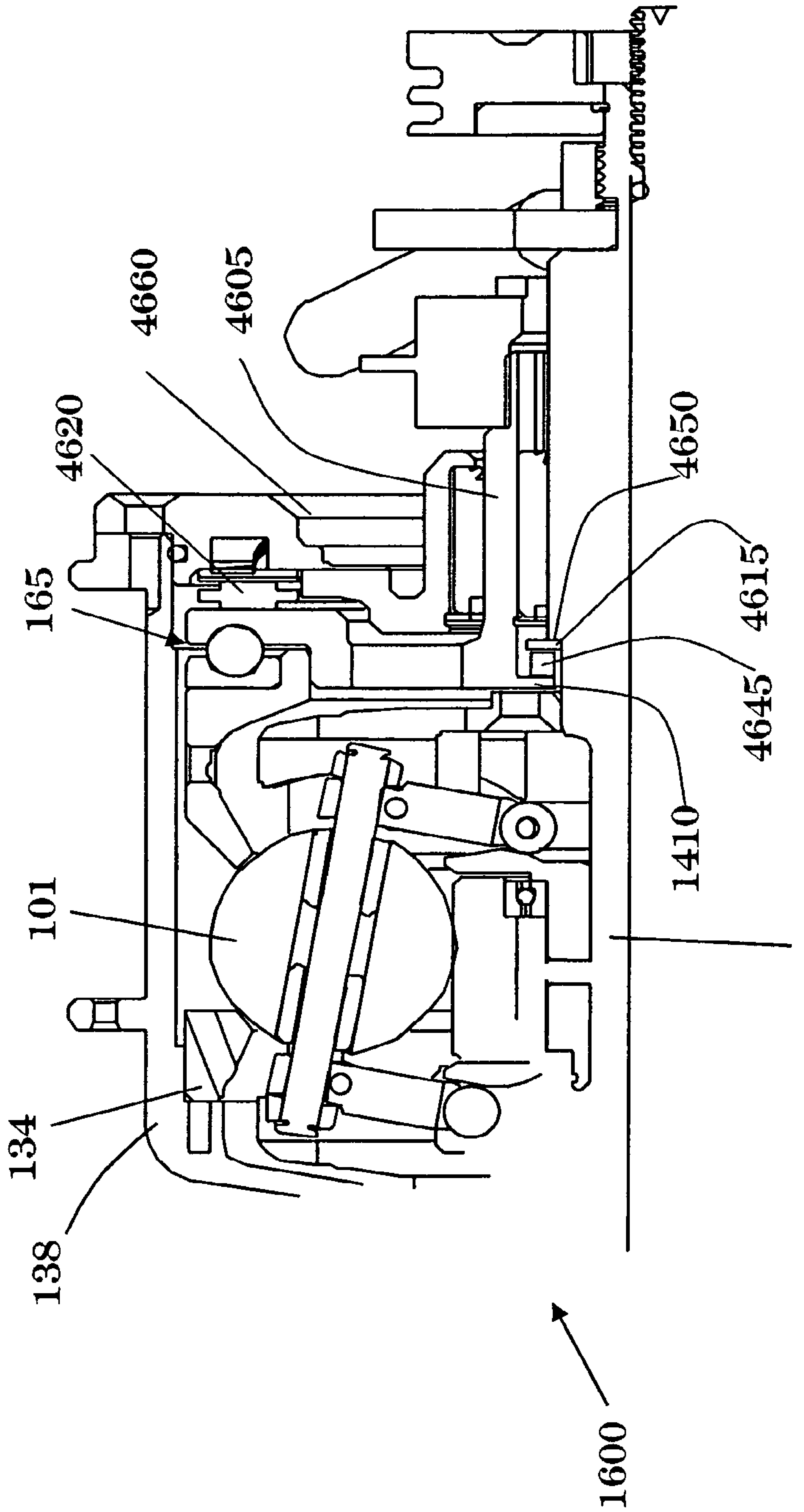

FIGURE 46

CONTINUOUSLY VARIABLE TRANSMISSION

RELATED APPLICATIONS

This application is a continuation of U.S. application Ser. No. 11/842,081, filed Aug. 20, 2007, which is a continuation of U.S. application Ser. No. 11/243,484, filed Oct. 4, 2005 and issued as U.S. Pat. No. 7,762,919 on Jul. 27, 2010, which claims the benefit of U.S. Provisional Application No. 60/616,399, filed on Oct. 5, 2004. Each of the above-identified applications is incorporated by reference in its entirety.

BACKGROUND OF THE INVENTION

1. Field of the Invention

The field of the invention relates generally to transmissions, and more particularly to continuously variable transmissions (CVTs).

2. Description of the Related Art

There are well-known ways to achieve continuously variable ratios of input speed to output speed. The mechanism for adjusting an input speed from an output speed in a CVT is known as a variator. In a belt-type CVT, the variator consists of two adjustable pulleys having a belt between them. The variator in a single cavity toroidal-type CVT has two partially toroidal transmission discs rotating about a shaft and two or more disc-shaped power rollers rotating on respective axes that are perpendicular to the shaft and clamped between the input and output transmission discs.

Embodiments of the invention disclosed here are of the spherical-type variator utilizing spherical speed adjusters (also known as power adjusters, balls, sphere gears or rollers) that each has a tiltable axis of rotation; the adjusters are distributed in a plane about a longitudinal axis of a CVT. The rollers are contacted on one side by an input disc and on the other side by an output disc, one or both of which apply a clamping contact force to the rollers for transmission of torque. The input disc applies input torque at an input rotational speed to the rollers. As the rollers rotate about their own axes, the rollers transmit the torque to the output disc. The input speed to output speed ratio is a function of the radii of the contact points of the input and output discs to the axes of the rollers. Tilting the axes of the rollers with respect to the axis of the variator adjusts the speed ratio.

SUMMARY OF INVENTION

One embodiment is a CVT. The CVT includes a central shaft and a variator. The variator includes an input disc, an output disc, a plurality of tiltable ball-leg assemblies, and an idler assembly. The input disc is rotatably mounted about the central shaft. Each of the plurality of tiltable ball-leg assemblies includes a ball, an axle, and at least two legs. The ball is rotatably mounted to the axle and contacts the input disk and the output disk. The legs are configured to control the tilt of the ball. The idler assembly is configured to control the radial position of the legs so as to thereby control the tilt of the ball. In one embodiment, the CVT is adapted for use in a bicycle.

In one embodiment, the variator includes a disk having a splined bore and a driver with splines. The splines of the driver couple to the splined bore of the disk.

In one embodiment, a shift rod extends through the central shaft and connects to the idler assembly. The shift rod actuates the idler assembly.

In one embodiment, a cam loader is positioned adjacent to the input disc and is configured to at least partly generate axial

force and transfer torque. In one embodiment, a cam loader is positioned adjacent to the output disc and is configured to at least partly generate axial force and transmit torque. In yet other embodiments, cam loaders are positioned adjacent to both the input disc and the output disc; the cam loaders are configured to at least partly generate axial force and transmit torque.

Another embodiment is a spacer for supporting and separating a cage of a CVT having a hub shell that at least partially encloses a variator. The spacer includes a scraper configured to scrape lubricant from a surface of the hub shell and direct the lubricant toward the inside of the variator. In one embodiment, the spacer includes passages configured to direct the flow of lubricant.

Another aspect of the invention relates to a torsion disc for a CVT. The torsion disc includes a spline bore about its central axis, an annular recess formed in the disc for receiving the race of a bearing, and a raised surface for supporting a torsion spring.

Yet another feature of the invention concerns a shaft for supporting certain components of a CVT. In some embodiments, the shaft has a splined flange, a central bore spanning from one end of the shaft to a point beyond the middle of the shaft, and one or more flanges for attaching to various components of the CVT. In one embodiment, flanges on the shaft are adapted to couple to stators of the CVT.

A different aspect of the inventive CVTs relates to an axial force generating system having a torsion spring coupled to a torsion disc and an input disc of the CVT. The axial force generating system may also include one or more load cam discs having ramps for energizing rollers, which are preferably located between the load cam disc and the input disc and/or output disc of the CVT.

Another feature of the invention is directed to an axle and axle-ball combination for a CVT. In some embodiments, the axle includes shoulder portions and a waist portion. The axle is configured to fit in a central bore of a traction roller of the CVT. In some embodiments, the bearing surface between the axle and the ball may be a journal bearing, a bushing, a Babbitt lining, or the axle itself. In other embodiments, the axle and ball utilize retained bearings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a cross-sectional view of one embodiment of a CVT.

FIG. **2** is a partially exploded cross-sectional view of the CVT of FIG. **1**.

FIG. **3** is a cross-sectional view of a second embodiment of a CVT.

FIG. **4** is a partially exploded cross-sectional view of the CVT of FIG. **3**.

FIG. **5*a*** is a side view of a splined input disc driver that can be used in a CVT.

FIG. **5*b*** is a front view of the disc driver of FIG. **5*a***.

FIG. **6*a*** is a side view of a splined input disc that can be used in a CVT.

FIG. **6*b*** is a front view of the splined input disc of FIG. **6*a***.

FIG. **7** is a cam roller disc that can be used with a CVT.

FIG. **8** is a stator that can be used with a CVT.

FIG. **9** is a perspective view of a scraping spacer that can be used with a CVT.

FIG. **10** is a cross-sectional view of a shifter assembly that can be used in a CVT.

FIG. **11** is a perspective view of a ball-leg assembly for use in a CVT.

FIG. **12** is a perspective view of a cage that can be used in a ball-type CVT.

FIG. **13** is a cross-sectional view of another embodiment of a CVT.

FIG. **14** is a perspective view of a bicycle hub incorporating an embodiment of a CVT.

FIG. **15** is a top elevational view of various assemblies of an embodiment of a CVT incorporated in the bicycle hub of FIG. **14**.

FIG. **16** is a partially exploded, perspective view of certain assemblies of the CVT of FIG. **15**.

FIG. **17** is a top elevational view of certain assemblies of the CVT of FIG. **15**.

FIG. **18** is a cross-sectional view along section A-A of the assemblies of FIG. **17**.

FIG. **19** is a perspective view of one embodiment of a shift cam assembly that can be used with the CVT of FIG. **15**.

FIG. **20** is a top elevational view of the shift cam assembly of FIG. **19**.

FIG. **21** is a cross-sectional view along section B-B of the shift cam assembly of FIG. **20**.

FIG. **22** is perspective view of a cage assembly that can be used with the CVT of FIG. **15**.

FIG. **23** is a front elevational view of the cage assembly of FIG. **22**.

FIG. **24** is a right side elevational view of the cage assembly of FIG. **22**.

FIG. **25** is a partially exploded, front elevational view of certain axial force generation components for the CVT of FIG. **15**.

FIG. **26** is a cross-sectional view along section C-C of the CVT components shown in FIG. **25**.

FIG. **27** is an exploded perspective view of a mating input shaft and torsion disc that can be used with the CVT of FIG. **15**.

FIG. **28** is a perspective view of the torsion disc of FIG. **27**.

FIG. **29** is a left side elevational view of the torsion disc of FIG. **28**.

FIG. **30** is a front elevation view of the torsion disc of FIG. **28**.

FIG. **31** is a right side elevational view of the torsion disc of FIG. **28**.

FIG. **32** is a cross-sectional view along section D-D of the torsion disc of FIG. **31**.

FIG. **33** is a perspective view of the input shaft of FIG. **27**.

FIG. **34** is a left side elevational view of the input shaft of FIG. **33**.

FIG. **35** is a top side elevational view of the input shaft of FIG. **33**.

FIG. **36** is a perspective view of a load cam disc that can be used with the CVT of FIG. **15**.

FIG. **37** is a top side elevational view of a ball and axle assembly that can be used with the CVT of FIG. **15**.

FIG. **38** is a cross-sectional view along section E-E of the ball and axle assembly of FIG. **37**.

FIG. **39** is a top elevational view of the bicycle hub of FIG. **14**.

FIG. **40** is a cross-sectional view along section F-F of the hub of FIG. **39** showing certain components of the bicycle hub of FIG. **14** and the CVT of FIG. **15**.

FIG. **41** is a perspective view of a main shaft that can be used with the CVT of FIG. **15**.

FIG. **42** is a top side elevational view of the main shaft of FIG. **41**.

FIG. **43** is a cross-section view along section G-G of the main shaft of FIG. **42**.

FIG. **44** is a top elevational view of an alternative embodiment of a CVT that can be used with the bicycle hub of FIG. **14**.

FIG. **45** is a cross-sectional view along section H-H of the CVT of FIG. **44**.

FIG. **46** is a cross-sectional view of a CVT that can be used with the bicycle hub of FIG. **14**.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

The CVT embodiments described here are generally of the type disclosed in U.S. Pat. Nos. 6,241,636, 6,419,608 and 6,689,012. The entire disclosure of each of these patents is hereby incorporated herein by reference.

FIG. **1** illustrates a spherical-type CVT **100** that can change input to output speed ratios. The CVT **100** has a central shaft **105** extending through the center of the CVT **100** and beyond two rear dropouts **10** of the frame of a bicycle. A first cap nut **106** and second cap nut **107**, each located at a corresponding end of the central shaft **105**, attach the central shaft **105** to the dropouts. Although this embodiment illustrates the CVT **100** for use on a bicycle, the CVT **100** can be implemented on any equipment that makes use of a transmission. For purposes of description, the central shaft **105** defines a longitudinal axis of the CVT that will serve as a reference point for describing the location and or motion of other components of the CVT. As used here, the terms “axial,” “axially,” “lateral,” “laterally,” refer to a position or direction that is coaxial or parallel with the longitudinal axis defined by the central shaft **105**. The terms “radial” and “radially” refer to locations or directions that extend perpendicularly from the longitudinal axis.

Referring to FIGS. **1** and **2**, the central shaft **105** provides radial and lateral support for a cage assembly **180**, an input assembly **155** and an output assembly **160**. In this embodiment the central shaft **105** includes a bore **199** that houses a shift rod **112**. As will be described later, the shift rod **112** actuates a speed ratio shift in the CVT **100**.

The CVT **100** includes a variator **140**. The variator **140** can be any mechanism adapted to change the ratio of input speed to output speed. In one embodiment, the variator **140** includes an input disc **110**, an output disc **134**, tiltable ball-leg assemblies **150** and an idler assembly **125**. The input disc **110** may be a disc mounted rotatably and coaxially about the central shaft **105**. At the radial outer edge of the input disc **110**, the disc extends at an angle to a point where it terminates at a contact surface **111**. In some embodiments, the contact surface **111** can be a separate structure, for example a ring that attaches to the input disc **110**, which would provide support for the contact surface **111**. The contact surface **111** may be threaded, or press fit, into the input disc **110** or it can be attached with any suitable fasteners or adhesives.

The output disc **134** can be a ring that attaches, by press fit or otherwise, to an output hub shell **138**. In some embodiments, the input disc **110** and the output disc **134** have support structures **113** that extend radially outward from contact surfaces **111** and that provide structural support to increase radial rigidity, to resist compliance of those parts under the axial force of the CVT **100**, and to allow axial force mechanisms to move radially outward, thereby reducing the length of the CVT **100**. The input disc **110** and the output disc **134** can have oil ports **136**, **135** to allow lubricant in the variator **140** to circulate through the CVT **100**.

The hub shell **138** in some embodiments is a cylindrical tube rotatable about the central shaft **105**. The hub shell **138** has an inside that houses most of the components of the CVT **100** and an outside adapted to connect to whatever component, equipment or vehicle uses the CVT. Here the outside of the hub shell **138** is configured to be implemented on a bicycle. However, the CVT **100** can be used in any machine where it is desirable to adjust rotational input and output speeds.

Referring to FIGS. **1**, **2**, **10** and **11** a CVT may include a ball-leg assembly **150** for transmitting torque from the input disc **110** to the output disc **134** and varying the ratio of input speed to output speed. In some embodiments, the ball-leg assembly **150** includes a ball **101**, a ball axle **102**, and legs **103**. The axle **102** can be a generally cylindrical shaft that extends through a bore formed through the center of the ball **101**. In some embodiments, the axle **102** interfaces with the surface of the bore in the ball **101** via needle or radial bearings that align the ball **101** on the axle **102**. The axle **102** extends beyond the sides of the ball **101** where the bore ends so that the legs **103** can actuate a shift in the position of the ball **101**. Where the axle **102** extends beyond the edge of the ball **101**, it couples to the radial outward end of the legs **103**. The legs **103** are radial extensions that tilt the ball axle **102**.

The axle **102** passes through a bore formed in the radially outward end of a leg **103**. In some embodiments, the leg **103** has chamfers where the bore for the axle **102** passes through the legs **103**, which provides for reduced stress concentration at the contact between the side of the leg **103** and the axle **102**. This reduced stress increases the capacity of the ball-leg assembly **150** to absorb shifting forces and torque reaction. The leg **103** can be positioned on the axle **102** by clip rings, such as e-rings, or can be press fit onto the axle **102**; however, any other type of fixation between the axle **102** and the leg **103** can be utilized. The ball-leg assembly **150** can also include leg rollers **151**, which are rolling elements attached to each end of a ball axle **102** and provide for rolling contact of the axle **102** as it is aligned by other parts of the CVT **100**. In some embodiments, the leg **103** has a cam wheel **152** at a radially inward end to help control the radial position of the leg **103**, which controls the tilt angle of the axle **102**. In yet other embodiments, the leg **103** couples to a stator wheel **1105** (see FIG. **11**) that allows the leg **103** to be guided and supported in the stators **800** (see FIG. **8**). As shown in FIG. **11**, the stator wheel **1105** may be angled relative to the longitudinal axis of the leg **103**. In some embodiments, the stator wheel **1105** is configured such that its central axis intersects with the center of the ball **101**.

Still referring to FIGS. **1**, **2**, **10** and **11**, in various embodiments the interface between the balls **101** and the axles **102** can be any of the bearings described in other embodiments below. However, the balls **101** are fixed to the axles in other embodiments and rotate with the balls **101**. In some such embodiments, bearings (not shown) are positioned between the axles **102** and the legs **103** such that the transverse forces acting on the axles **102** are reacted by the legs **103** as well as, or alternatively, the cage (described in various embodiments below). In some such embodiments, the bearing positioned between the axles **102** and the legs **103** are radial bearings (balls or needles), journal bearings or any other type of bearings or suitable mechanism or means.

With reference to FIGS. **1**, **2**, **3**, **4** and **10**, the idler assembly **125** will now be described. In some embodiments, the idler assembly **125** includes an idler **126**, cam discs **127**, and idler bearings **129**. The idler **126** is a generally cylindrical tube. The idler **126** has a generally constant outer diameter; however, in other embodiments the outer diameter is not constant. The outer diameter may be smaller at the center portion than at the ends, or may be larger at the center and smaller at the ends. In other embodiments, the outer diameter is larger at one end than at the other and the change between the two ends may be linear or non-linear depending on shift speed and torque requirements.

The cam discs **127** are positioned on either or both ends of the idler **126** and interact with the cam wheels **152** to actuate the legs **103**. The cam discs **127** are convex in the illustrated embodiment, but can be of any shape that produces a desired motion of the legs **103**. In some embodiments, the cam discs **127** are configured such that their axial position controls the radial position of the legs **103**, which governs the angle of tilt of the axles **102**.

In some embodiments, the radial inner diameter of the cam discs **127** extends axially toward one another to attach one cam disc **127** to the other cam disc **127**. Here, a cam extension **128** forms a cylinder about the central shaft **105**. The cam extension **128** extends from one cam disc **127** to the other cam disc **127** and is held in place there by a clip ring, a nut, or some other suitable fastener. In some embodiments, one or both of the cam discs **127** are threaded onto the cam disc extension **128** to fix them in place. In the illustrated embodiment, the convex curve of the cam disc **127** extends axially away from the axial center of the idler assembly **125** to a local maximum, then radially outward, and back axially inward toward the axial center of the idler assembly **125**. This cam profile reduces binding that can occur during shifting of the idler assembly **125** at the axial extremes. Other cam shapes can be used as well.

In the embodiment of FIG. **1**, a shift rod **112** actuates a transmission ratio shift of the CVT **100**. The shift rod **112**, coaxially located inside the bore **199** of the central shaft **105**, is an elongated rod having a threaded end **109** that extends out one side of the central shaft **105** and beyond the cap nut **107**. The other end of the shift rod **112** extends into the idler assembly **125** where it contains a shift pin **114**, which mounts generally transversely in the shift rod **112**. The shift pin **114** engages the idler assembly **125** so that the shift rod **112** can control the axial position of the idler assembly **125**. A lead screw assembly **115** controls the axial position of the shift rod **112** within the central shaft **105**. In some embodiments, the lead screw assembly **125** includes a shift actuator **117**, which may be a pulley having a set of tether threads **118** on its outer diameter with threads on a portion of its inner diameter to engage the shift rod **112**. The lead screw assembly **115** may be held in its axial position on the central shaft **105** by any means, and here is held in place by a pulley snap ring **116**. The tether threads **118** engage a shift tether (not shown). In some embodiments, the shift tether is a standard shift cable, while in other embodiments the shift tether can be any tether capable of supporting tension and thereby rotating the shift pulley **117**.

Referring to FIGS. **1** and **2**, the input assembly **155** allows torque transfer into the variator **140**. The input assembly **155** has a sprocket **156** that converts linear motion from a chain (not shown) into rotational motion. Although a sprocket is used here, other embodiments of the CVT **100** may use a pulley that accepts motion from a belt, for example. The sprocket **156** transmits torque to an axial force generating mechanism, which in the illustrated embodiment is a cam loader **154** that transmits the torque to the input disc **110**. The cam loader **154** includes a cam disc **157**, a load disc **158** and a set of cam rollers **159**. The cam loader **154** transmits torque from the sprocket **156** to the input disc **110** and also generates an axial force that resolves into the contact force for the input disc **110**, the balls **101**, the idler **126** and the output disc **134**. The axial force is generally proportional to the amount of torque applied to the cam loader **154**. In some embodiments, the sprocket **156** applies torque to the cam disc **157** via a one-way clutch (detail not shown) that acts as a coasting mechanism when the hub **138** spins but the sprocket **156** is not supplying torque. In some embodiments, the load disc **158** may be integral as a single piece with the input disc **157**. In other embodiments, the cam loader **154** may be integral with the output disc **134**.

In FIGS. **1** and **2**, the internal components of the CVT **100** are contained within the hub shell **138** by an end cap **160**. The end cap **160** is a generally flat disc that attaches to the open end of the hub shell **138** and has a bore through the center to allow passage of the cam disc **157**, the central shaft **105** and the shift rod **112**. The end cap **160** attaches to the hub shell **138** and serves to react the axial force created by the cam loader **154**. The end cap **160** can be made of any material capable of reacting the axial force such as for example, aluminum, titanium, steel, or high strength thermoplastics or thermoset plastics. The end cap **160** fastens to the hub shell **138** by fasteners (not shown); however, the end cap **160** can also thread into, or can otherwise be attached to, the hub shell **138**. The end cap **160** has a groove formed about a radius on its side facing the cam loader **154** that houses a preloader **161**. The preloader **161** can be a spring that provides and an initial clamp force at very low torque levels. The preloader **161** can be any device capable of supplying an initial force to the cam loader **154**, and thereby to the input disc **134**, such as a spring, or a resilient material like an o-ring. The preloader **161** can be a wave-spring as such springs can have high spring constants and maintain a high level of resiliency over their lifetimes. Here the preloader **161** is loaded by a thrust washer **162** and a thrust bearing **163** directly to the end cap **160**. In this embodiment, the thrust washer **162** is a typical ring washer that covers the groove of the preloader **161** and provides a thrust race for the thrust bearing **163**. The thrust bearing **163** may be a needle thrust bearing that has a high level of thrust capacity, improves structural rigidity, and reduces tolerance requirements and cost when compared to combination thrust radial bearings; however, any other type of thrust bearing or combination bearing can be used. In certain embodiments, the thrust bearing **163** is a ball thrust bearing. The axial force developed by the cam loader **154** is reacted through the thrust bearing **163** and the thrust washer **162** to the end cap **160**. The end cap **160** attaches to the hub shell **138** to complete the structure of the CVT **100**.

In FIGS. **1** and **2**, a cam disc bearing **172** holds the cam disc **157** in radial position with respect to the central shaft **105**, while an end cap bearing **173** maintains the radial alignment between the cam disc **157** and the inner diameter of the end cap **160**. Here the cam disc bearing **172** and the end cap bearing **173** are needle roller bearings; however, other types of radial bearings can be used as well. The use of needle roller bearings allow increased axial float and accommodates binding moments developed by the rider and the sprocket **156**. In other embodiments of the CVT **100** or any other embodiment described herein, each of or either of the can disc bearing **172** and the end cap bearing **173** can also be replaced by a complimentary pair of combination radial-thrust bearings. In such embodiments, the radial thrust bearings provide not only the radial support but also are capable of absorbing thrust, which can aid and at least partially unload the thrust bearing **163**.

Still referring to FIGS. **1** and **2**, an axle **142**, being a support member mounted coaxially about the central shaft **105** and held between the central shaft **105** and the inner diameter of the closed end of the hub shell **138**, holds the hub shell **138** in radial alignment with respect to the central shaft **105**. The axle **142** is fixed in its angular alignment with the central shaft **105**. Here a key **144** fixes the axle **142** in its angular alignment, but the fixation can be by any means known to those of skill in the relevant technology. A radial hub bearing **145** fits between the axle **142** and the inner diameter of the hub shell **138** to maintain the radial position and axial alignment of the hub shell **138**. The hub bearing **145** is held in place by an encapsulating axle cap **143**. The axle cap **143** is a disc having a central bore that fits around central shaft **105** and here attaches to the hub shell **138** with fasteners **147**. A hub thrust bearing **146** fits between the hub shell **138** and the cage **189** to maintain the axial positioning of the cage **189** and the hub shell **138**.

FIGS. **3**, **4** and **10** illustrate a CVT **300**, which is an alternative embodiment of the CVT **100** described above. Many of the components are similar between the CVT **100** embodiments described above and that of the present figures. Here, the angles of the input and output discs **310**, **334** respectively are decreased to allow for greater strength to withstand axial forces and to reduce the overall radial diameter of the CVT **300**. This embodiment shows an alternate shifting mechanism, where the lead screw mechanism to actuate axial movement of the idler assembly **325** is formed on the shift rod **312**. The lead screw assembly is a set of lead threads **313** formed on the end of the shift rod **312** that is within or near the idler assembly **325**. One or more idler assembly pins **314** extend radially from the cam disc extensions **328** into the lead threads **313** and move axially as the shift rod **312** rotates.

In the illustrated embodiment, the idler **326** does not have a constant outer diameter, but rather has an outer diameter that increases at the ends of the idler **326**. This allows the idler **326** to resist forces of the idler **326** that are developed through the dynamic contact forces and spinning contact that tend to drive the idler **326** axially away from a center position. However, this is merely an example and the outer diameter of the idler **326** can be varied in any manner a designer desires in order to react the spin forces felt by the idler **326** and to aid in shifting of the CVT **300**.

Referring now to FIGS. **5***a*, **5***b*, **6***a*, and **6***b*, a two part disc is made up of a splined disc **600** and a disc driver **500**. The disc driver **500** and the splined disc **600** fit together through splines **510** formed on the disc driver **500** and a splined bore **610** formed in the splined disc **600**. The splines **510** fit within the splined bore **610** so that the disc driver **500** and the splined disc **600** form a disc for use in the CVT **100**, CVT **300**, or any other spherical CVT. The splined disc **600** provides for compliance in the system to allow the variator **140**, **340** to find a radial equilibrium position so as to reduce sensitivity to manufacturing tolerances of the components of a variator **140**, **340**.

FIG. **7** illustrates a cam disc **700** that can be used in the CVT **100**, CVT **300**, other spherical CVTs or any other type of CVT. The cam disc **700** has cam channels **710** formed in its radial outer edge. The cam channels **710** house a set of cam rollers (not shown) which in this embodiment are spheres (such as bearing balls) but can be any other shape that combines with the shape of the cam channel **710** to convert torque into torque and axial force components to moderate the axial force applied to the variator **140**, **340** in an amount proportional to the torque applied to the CVT. Other such shapes include cylindrical rollers, barreled rollers, asymmetrical rollers or any other shape. The material used for the cam disc channels **710** in many embodiments is preferably strong enough to resist excessive or permanent deformation at the loads that the cam disc **700** will experience. Special hardening may be needed in high torque applications. In some embodiments, the cam disc channels **710** are made of carbon steel hardened to Rockwell hardness values above 40 HRC. The efficiency of the operation of the cam loader (**154** of FIG. **1**, or any other type of cam loader) can be affected by the hardness value, typically by increasing the hardness to increase the efficiency; however, high hardening can lead to brittleness in the cam loading components and can incur higher cost as well. In some embodiments, the hardness is above 50 HRC, while in other embodiments the hardness is above 55 HRC, above 60 HRC and above 65 HRC.

FIG. **7** shows an embodiment of a conformal cam. That is, the shape of the cam channel **710** conforms to the shape of the cam rollers. Since the channel **710** conforms to the roller, the channel **710** functions as a bearing roller retainer and the requirement of a cage element is removed. The embodiment of FIG. **7** is a single direction cam disc **700**; however, the cam disc can be a bidirectional cam as in the CVT **1300** (see FIG. **13**). Eliminating the need for a bearing roller retainer simplifies the design of the CVT. A conformal cam channel **710** also allows the contact stress between the bearing roller and the channel **710** to be reduced, allowing for reduced bearing roller size and/or count, or for greater material choice flexibility.

FIG. **8** illustrates a cage disc **800** used to form the rigid support structure of the cage **189** of the variators **140**, **340** in spherical CVTs **100**, **300** (and other types). The cage disc **800** is shaped to guide the legs **103** as they move radially inward and outward during shifting. The cage disc **800** also provides the angular alignment of the axles **102**. In some embodiments the corresponding grooves of two cage discs **800** for a respective axle **102** are offset slightly in the angular direction to reduce shift forces in the variators **140** and **340**.

Legs **103** are guided by slots in the stators. Leg rollers **151** on the legs **103** follow a circular profile in the stators. The leg rollers **151** generally provide a translational reaction point to counteract translational forces imposed by shift forces or traction contact spin forces. The legs **103** as well as its respective leg rollers **151** move in planar motion when the CVT ratio is changed and thus trace out a circular envelope which is centered about the ball **101**. Since the leg rollers **151** are offset from the center of the leg **103**, the leg rollers **151** trace out an envelope that is similarly offset. To create a compatible profile on each stator to match the planar motion of the leg rollers **151**, a circular cut is required that is offset from the groove center by the same amount that the roller is offset in each leg **103**. This circular cut can be done with a rotary saw cutter; however, it requires an individual cut at each groove. Since the cuts are independent, there is a probability of tolerance variation from one groove to the next in a single stator, in addition to variation between stators. A method to eliminate this extra machining step is to provide a single profile that can be generated by a lath turning operation. A toroidal-shaped lathe cut can produce this single profile in one turning operation. The center of the toroidal cut is adjusted away from the center of the ball **101** position in a radial direction to compensate for offset of the leg rollers **103**.

Referring now to FIGS. **1**, **9** and **12**, an alternative embodiment of a cage assembly **1200** is illustrated implementing a lubrication enhancing lubricating spacer **900** for use with some CVTs where spacers **1210** support and space apart two cage discs **1220**. In the illustrated embodiment, the support structure for the power transmission elements, in this case the cage **389**, is formed by attaching input and output side cage discs **1220** to a plurality of spacers **1210**, including one or more lubricating spacers **900** with cage fasteners **1230**. In this embodiment, the cage fasteners **1230** are screws but they can be any type of fastener or fastening method. The lubricating spacer **900** has a scraper **910** for scraping lubricant from the surface of the hub shell **138** and directing that lubricant back toward the center elements of the variator **140** or **340**. The lubricating spacer **900** of some embodiments also has passages **920** to help direct the flow of lubricant to the areas that most utilize it. In some embodiments, a portion of the spacer **900** between the passages **920** forms a raised wedge **925** that directs the flow of lubricant towards the passages **920**. The scraper **910** may be integral with the spacer **900** or may be separate and made of a material different from the material of the scraper **910**, including but not limited to rubber to enhance scraping of lubricant from the hub shell **138**. The ends of the spacers **1210** and the lubricating spacers **900** terminate in flange-like bases **1240** that extend perpendicularly to form a surface for mating with the cage discs **1220**. The bases **1240** of the illustrated embodiment are generally flat on the side facing the cage discs **1240** but are rounded on the side facing the balls **101** so as to form the curved surface described above that the leg rollers **151** ride on. The bases **1240** also form the channel in which the legs **103** ride throughout their travel.

An embodiment of a lubrication system and method will now be described with reference to FIGS. **3**, **9**, and **10**. As the balls **101** spin, lubricant tends to flow toward the equators of the balls **101**, and the lubricant is then sprayed out against the hub shell **138**. Some lubricant does not fall on the internal wall of the hub shell **138** having the largest diameter; however, centrifugal force makes this lubricant flow toward the largest inside diameter of the hub shell **138**. The scraper **910** is positioned vertically so that it removes lubricant that accumulates on the inside of the hub shell **138**. Gravity pulls the lubricant down each side of V-shaped wedge **925** and into the passages **920**. The spacer **900** is placed such that the inner radial end of the passages **920** end in the vicinity of the cam discs **127** and the idler **126**. In this manner the idler **126** and the cam discs **127** receive lubrication circulating in the hub shell **138**. In one embodiment, the scraper **910** is sized to clear the hub shell **138** by about 30 thousandths of an inch. Of course, depending on different applications, the clearance could be greater or smaller.

As shown in FIGS. **3** and **10**, a cam disc **127** can be configured so that its side facing the idler **226** is angled in order to receive lubricant falling from the passages **920** and direct the lubricant toward the space between the cam disc **127** and the idler **226**. After lubricant flows onto the idler **226**, the lubricant flows toward the largest diameter of the idler **226**, where some of the lubricant is sprayed at the axles **102**. Some of the lubricant falls from the passages **920** onto the idler **226**. This lubricant lubricates the idler **226** as well as the contact patch between the balls **101** and the idler **226**. Due to the inclines on each side of the idler **226**, some of the lubricant flows centrifugally out toward the edges of the idler **226**, where it then sprays out radially.

Referring to FIGS. **1**, **3** and **10**, in some embodiments, lubricant sprayed from the idler **126**, **226** towards the axle **102** falls on grooves **345**, which receive the lubricant and pump it inside the ball **101**. Some of the lubricant also falls on the contact surface **111** where the input disc **110** and output disc **134** contact the balls **101**. As the lubricant exits on one side of the ball **101**, the lubricant flows toward the equator of the balls **101** under centrifugal force. Some of this lubricant contacts the input disc **110** and ball **101** contact surface **111** and then flows toward the equator of the ball **101**. Some of the lubricant flows out radially along a side of the output disc **134** facing away from the balls **101**. In some embodiments, the input disc **110** and/or output disc **134** are provided with lubrication ports **136** and **135**, respectively. The lubrication ports **135**, **136** direct the lubrication toward the largest inside diameter of the hub shell **138**.

FIG. **13** illustrates an embodiment of a CVT **1300** having two cam-loaders **1354** that share the generation and distribution of axial force in the CVT **1300**. Here, the cam loaders

1354 are positioned adjacent to the input disc **1310** and the output disc **1334**. The CVT **1300** illustrates how torque can be supplied either via the input disc **1310** and out through the output disc **1334** or reversed so that torque is input through the output disc **1334** and output through the input disc **1310**.

FIG. **14** depicts a bicycle hub **1400** configured to incorporate inventive features of embodiments of the CVTs described here. Several components of the hub **1400** are the same as components described above; hence, further description of such components will be limited. The hub **1400** includes a hub shell **138** that couples to a hub cap **1460**. In some embodiments, the hub **1400** also includes an end cap **1410** that seals the end of the hub shell **138** opposite the hub cap **1460**. The hub shell **138**, the hub cap **1460**, and the end cap **1410** are preferably made of materials that provide structural strength and rigidity. Such materials include, for example, steel, aluminum, magnesium, high-strength plastics, etc. In some embodiments, depending on the specific requirements of a given application of the technology, other materials might be appropriate. For example, the hub shell **138** may be made from composites, thermo plastics, thermoset plastics, etc.

Referring now to FIG. **14**, the illustrated hub **1400** houses in its interior embodiments of the CVTs presented herein. A main shaft **105** supports the hub **1400** and provides for attachment to the dropouts **10** of a bicycle or other vehicle or equipment. The main shaft **105** of this embodiment is described in further detail with reference to FIGS. **41-43**. In some embodiments, as illustrated in FIGS. **15-18**, a CVT **1500** includes a shifting mechanism that incorporates a rod **112** with a threaded end **109**. Nuts **106** and **107** lock the dropouts **10** to the main shaft **105**. In the embodiment of FIG. **14**, the hub **1400** includes a freewheel **1420** that is operationally coupled to an input shaft (see FIG. **33** and FIG. **40**) for transferring a torque input into the CVT **1500**. It should be noted that although various embodiments and features of the CVTs described here are discussed with reference to a bicycle application, through readily recognizable modifications the CVTs and features thereof can be used in any vehicle, machine or device that uses a transmission.

With reference to FIGS. **15** and **16**, in one embodiment the CVT **1500** has an input disc **1545** for transferring torque to a set of spherical traction rollers (here shown as balls **101**). FIG. **16** is a partially exploded view of the CVT **1500**. The balls **101** transfer the torque to an output disc **1560**. One ball **101** is illustrated in this embodiment to provide clarity in illustrating the various features of the CVT **1500**, however, various embodiments of the CVT employ anywhere from 2 to 16 balls **101** or more depending on the torque, weight and size requirements of each particular application. Different embodiments use either 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16 or more balls **101**. An idler **1526**, mounted coaxially about the main shaft **105**, contacts and provides support for the balls **101** and maintains their radial position about the main shaft **105**. The input disc **1545** of some embodiments, has lubrication ports **1590** to facilitate circulation of lubricant in the CVT **1500**.

Referring additionally to FIGS. **37-38**, the ball **101** spins on an axle **3702**. Legs **103** and shift cams **1527** cooperate to function as levers that actuate a shift in the position of the axle **3702**, which shift results in a tilting of the ball **101** and, thereby, a shift in the transmission ratio as already explained above. A cage **1589** (see FIGS. **22-24**) provides for support and alignment of the legs **103** as the shift cams **1527** actuate a radial motion of the legs **103**. In one embodiment, the cage includes stators **1586** and **1587** that are coupled by stator spacers **1555**. In other embodiments, other cages **180**, **389**, **1200** are employed.

Referring additionally to FIGS. **41-43**, in the illustrated embodiment, the cage **1589** mounts coaxially and nonrotatably about the main shaft **105**. The stator **1586** rigidly attaches to a flange **4206** of the main shaft **105** in this embodiment. An additional flange **1610** holds the stator **1587** in place. A key **1606** couples the flange **1610** to the main shaft **105**, which has a key seat **1608** for receiving the key **1606**. Of course, the person of ordinary skill in the relevant technology will readily recognize that there are many equivalent and alternative methods for coupling the main shaft **105** to the flange **1610**, or coupling the stators **1586**, **1587** to the flanges **1620**, **4206**. In certain embodiments, the main shaft **105** includes a shoulder **4310** that serves to axially position and constrain the flange **1610**.

The end cap **1410** mounts on a radial bearing **1575**, which itself mounts over the flange **1610**. In one embodiment, the radial bearing **1575** is an angular contact bearing that supports loads from ground reaction and radially aligns the hub shell **138** to the main shaft **105**. In some embodiments, the hub **1400** includes seals at one or both ends of the main shaft **105**. For example, here the hub **1400** has a seal **1580** at the end where the hub shell **138** and end cap **1410** couple together. Additionally, in order to provide an axial force preload on the output side and to maintain axial position of the hub shell **138**, the hub **1400** may include spacers **1570** and a needle thrust bearing (not shown) between the stator **1587** and the radial bearing **1575**. The spacers **1570** mount coaxially about the flange **1610**. In some embodiments, the needle thrust bearing may not used, and in such cases the radial bearing **1575** may be an angular contact bearing adapted to handle thrust loads. The person of ordinary skill in the relevant technology will readily recognize alternative means to provide the function of carrying radial and thrust loads that the spacers **1570**, needle thrust bearing, and radial bearing provide.

Still referring to FIGS. **14**, **15** and **16**, in the embodiment illustrated, a variator **1500** for the hub **1400** includes an input shaft **1505** that operationally couples at one end to a torsion disc **1525**. The other end of the input shaft **1505** operationally couples to the freewheel **1420** via a freewheel carrier **1510**. The torsion disc **1525** is configured to transfer torque to a load cam disc **1530** having ramps **3610** (see FIG. **36**). The load cam disc **1530** transfers torque and axial force to a set of rollers **2504** (see FIG. **25**), which act upon a second load cam disc **1540**. The input disc **1545** couples to the second load cam disc **1540** to receive torque and axial force inputs. In some embodiments, the rollers **2504** are held in place by a roller cage **1535**.

As is well known, many traction-type CVTs utilize a clamping mechanism to prevent slippage between the balls **101** and the input disc **1545** and/or output disc **1560** when transmitting certain levels of torque. Provision of a clamping mechanism is sometimes referred to here as generating an axial force, or providing an axial force generator. The configuration described above of the load cam disc **1530** acting in concert with the load cam **1540** through the rollers **2504** is one such axial force generating mechanism. However, as the axial force generating device or sub-assembly generates axial force in a CVT, reaction forces are also produced that are reacted in the CVT itself in some embodiments. Referring additionally to FIGS. **25** and **26**, in the embodiment illustrated of the CVT **1500**, the reaction forces are reacted at least in part by a thrust bearing having first and second races **1602** and **1603**, respectively. In the illustrated embodiment, the bearing elements are not shown but may be balls, rollers, barreled rollers, asymmetrical rollers or any other type of rollers. Additionally, in some embodiments, one or both of the races **1602** are made of various bearing race materials such as steel, bearing steel, ceramic or any other material used for bearing races. The first race **1602** butts up against the torsion disc **1525**, and the second race **1603** butts up against the hub cap **1460**. The hub cap **1460** of the illustrated embodiment helps to absorb the reaction forces that the axial force mechanism generates. In some embodiments, axial force generation involves additionally providing preloaders, such as one or more of an axial spring such as a wave spring **1515** or a torsion spring **2502** (see description below for FIG. **25**).

Referring to FIGS. **15-18**, **22-24** and **43**, certain subassemblies of the CVT **1500** are illustrated. The stator **1586** mounts on a shoulder **4208** of the main shaft **105** and butts up against the flange **4206** of the main shaft **105**. The stator **1587** mounts on a shoulder **1810** of the flange **1610**. Here, screws (not shown) attach the flange **4206** to the stator **1586** and attach the flange **1610** to the stator **1587**, however, in other embodiments the stator **1587** threads onto the shoulder **1810**, although the stator **1587** can be attached by any method or means to the shoulder **1810**. Because the flanges **1610** and **4206** are nonrotatably fixed to main shaft **105**, the cage **1589** made of the stators **1586** and **1587**, among other things, attaches nonrotatably in this embodiment to the main shaft **105**. The stator spacers **1555** provide additional structural strength and rigidity to the cage **1589**. Additionally, the stator spacers **1555** aid in implementing the accurate axial spacing between stators **1586** and **1587**. The stators **1586** and **1587** guide and support the legs **103** and axles **3702** through guide grooves **2202**.

Referring now to FIGS. **15-21**, **37**, **38**, the ball **101** spins about the axle **3702** and is in contact with an idler **1526**. Bearings **1829**, mounted coaxially about the main shaft **105**, support the idler **1526** in its radial position, which bearings **1829** may be separate from or integral with the idler **1526**. A shift pin **114**, controlled by the shift rod **112**, actuates an axial movement of the shift cams **1527**. The shift cams **1527** in turn actuate legs **103**, functionally resulting in the application of a lever or pivoting action upon the axle **3702** of the ball **101**. In some embodiments, the CVT **1500** includes a retainer **1804** that keeps the shift pin **114** from interfering with the idler **1526**. The retainer **1804** can be a ring made of plastic, metal, or other suitable material. The retainer **1804** fits between the bearings **1829** and mounts coaxially about a shift cam extension **1528**.

FIGS. **19-21** show one embodiment of the shift cams **1527** for the illustrated CVT **1500**. Each shift cam disc **1572** has a profile **2110** along which the legs **103** ride. Here the profile **2110** has a generally convex shape. Usually the shape of the profile **2110** is determined by the desired motion of the legs **103**, which ultimately affects the shift performance of the CVT **1500**. Further discussion of shift cam profiles is provided below. As shown, one of the shift cam discs **1527** has an extension **1528** that mounts about the main shaft **105**. The extension **1528** of the illustrated embodiment is sufficiently long to extend beyond the idler **1526** and couple to the other shift cam disc **1527**. Coupling here is provided by a slip-fit and a clip. However, in other embodiments, the shift cams **1527** can be fastened to each other by threads, screws, interference fit, or any other connection method. In some embodiments, the extension **1528** is provided as an extension from each shift cam **1527**. The shift pin **114** fits in a hole **1910** that goes through the extension **1528**. In some embodiments, the shift cams **1527** have orifices **1920** to improve lubrication flow through the idler bearings **1829**. In some embodiments the idler bearings **1829** are press fit onto the extension **1528**. In such embodiments, the orifices **1920** aid in removing the idler bearings **1829** from the extension **1528** by allowing a tool to pass through the shift cams **1527** and push the idler bearings **1829** off the extension **1528**. In certain embodiments, the idler bearings **1829** are angle contact bearings, while in other embodiments they are radial bearings or thrust bearings or any other type of bearing. Many materials are suitable for making the shift cams **1527**. For example, some embodiments utilize metals such as steel, aluminum, and magnesium, while other embodiments utilize other materials, such as composites, plastics, and ceramics, which depend on the conditions of each specific application.

The illustrated shift cams **1527** are one embodiment of a shift cam profile **2110** having a generally convex shape. Shift cam profiles usually vary according to the location of the contact point between the idler **1526** and the ball-leg assembly **1670** (see FIG. **16**) as well as the amount of relative axial motion between the ball **101** and the idler **1526**.

Referring now to the embodiment illustrated in FIGS. **16**, and **18-21**, the profile of shift cams **1527** is such that axial translation of the idler **1526** relative to the ball **101** is proportional to the change of the angle of the axis of the ball **101**. The angle of the axis of the ball **101** is referred to herein as "gamma." The applicant has discovered that controlling the axial translation of the idler **1526** relative to the change in gamma influences CVT ratio control forces. For example, in the illustrated CVT **1500**, if the axial translation of the idler **1526** is linearly proportional to a change in gamma, the normal force at the shift cams **1527** and ball-leg interface is generally parallel to the axle **3702**. This enables an efficient transfer of horizontal shift forces to a shift moment about the ball-leg assembly **1670**.

A linear relation between idler translation and gamma is given as idler translation is the mathematical product of the radius of the balls **101**, the gamma angle and RSF (i.e., idler translation=ball radius*gamma angle*RSF), where RSF is a roll-slide factor. RSF describes the transverse creep rate between the ball **101** and the idler **126**. As used here, "creep" is the discrete local motion of a body relative to another. In traction drives, the transfer of power from a driving element to a driven element via a traction interface requires creep. Usually, creep in the direction of power transfer is referred to as "creep in the rolling direction." Sometimes the driving and driven elements experience creep in a direction orthogonal to the power transfer direction, in such a case this component of creep is referred to as "transverse creep." During CVT operation, the ball **101** and idler **1526** roll on each other. When the idler is shifted axially (i.e., orthogonal to the rolling direction), transverse creep is imposed between the idler **1526** and the ball **101**. An RSF equal to 1.0 indicates pure rolling. At RSF values less than 1.0, the idler **1526** translates slower than the ball **101** rotates. At RSF values greater than 1.0, the idler **1526** translates faster than the ball **101** rotates.

Still referring to the embodiments illustrated in FIGS. **16**, and **18-21**, the applicant has devised a process for layout of the cam profile for any variation of transverse creep and/or location of the interface between the idler **1526** and the ball-leg assembly **1570**. This process generates different cam profiles and aids in determining the effects on shift forces and shifter displacement. In one embodiment, the process involves the use of parametric equations to define a two-dimensional datum curve that has the desired cam profile. The curve is then used to generate models of the shift cams **127**. In one embodiment of the process, the parametric equations of the datum curve are as follows:

theta=2*GAMMA_MAX*t−GAMMA_MAX

x=LEG*sin(theta)−0.5*BALL_DIA*RSF*theta*pi/180+0.5*ARM*cos(theta)

y=LEG*cos(theta)−0.5*ARM*sin(theta)

z=0

The angle theta varies from minimum gamma (which in some embodiments is −20 degrees) to maximum gamma (which in some embodiments is +20 degrees). GAMMA_MAX is the maximum gamma. The parametric range variable "t" varies from 0 to 1. Here "x" and "y" are the center point of the cam wheel **152** (see FIG. **1**). The equations for x and y are parametric. "LEG" and "ARM" define the position of the interface between the ball-leg assembly **1670** and the idler **1526** and shift cams **1527**. More specifically, LEG is the perpendicular distance between the axis of the ball axle **3702** of a ball-leg assembly **1670** to a line that passes through the centers of the two corresponding cam wheels **152** of that ball-leg assembly **1570**, which is parallel to the ball axle **3702**. ARM is the distance between centers of the cam wheels **152** of a ball-leg-assembly **1670**.

RSF values above zero are preferred. The CVT **100** demonstrates an application of RSF equal to about 1.4. Applicant discovered that an RSF of zero dramatically increases the force required to shift the CVT. Usually, RSF values above 1.0 and less than 2.5 are preferred.

Still referring to the embodiments illustrated in FIGS. **16**, and **18-21**, in the illustrated embodiment of a CVT **100**, there is a maximum RSF for a maximum gamma angle. For example, for gamma equals to +20 degrees an RSF of about 1.6 is the maximum. RSF further depends on the size of the ball **101** and the size of the idler **1526**, as well as the location of the cam wheel **152**.

In terms of energy input to shift the CVT, the energy can be input as a large displacement and a small force (giving a large RSF) or a small displacement and a large force (giving a small RSF). For a given CVT there is a maximum allowable shift force and there is also a maximum allowable displacement. Hence, a trade off offers designers various design options to be made for any particular application. An RSF greater than zero reduces the required shift force by increasing the axial displacement necessary to achieve a desired shift ratio. A maximum displacement is determined by limits of the particular shifting mechanism, such as a grip or trigger shift in some embodiments, which in some embodiments can also be affected or alternatively affected by the package limits for the CVT **100**.

Energy per time is another factor. Shift rates for a given application may require a certain level of force or displacement to achieve a shift rate depending on the power source utilized to actuate the shift mechanism. For example, in certain applications using an electric motor to shift the CVT, a motor having a high speed at low torque would be preferred in some instances. Since the power source is biased toward speed, the RSF bias would be toward displacement. In other applications using hydraulic shifting, high pressure at low flow may be more suitable than low pressure at high flow. Hence, one would choose a lower RSF to suit the power source depending on the application.

Idler translation being linearly related to gamma is not the only desired relation. Hence, for example, if it is desired that the idler translation be linearly proportional to CVT ratio, then the RSF factor is made a function of gamma angle or CVT ratio so that the relation between idler position and CVT ratio is linearly proportional. This is a desirable feature for some types of control schemes.

FIGS. **22-24** show one example of a cage **1589** that can be used in the CVT **1500**. The illustrated cage **1589** has two stators **1586** and **1587** coupled to each other by a set of stator spacers **1555** (only one is shown for clarity). The stator spacers **1555** in this embodiment fasten to the outer periphery of the stators **1586** and **1587**. Here screws attach the spacers **1555** to the stators **1586** and **1587**. However, the stators **1586** and **1587** and the spacers **1555** can be configured for other means of attachment, such as press fitting, threading, or any other method or means. In some embodiments, one end of the spacers **1555** is permanently affixed to one of the stators **1586** or **1587**. In some embodiments, the spacers **1555** are made of a material that provides structural rigidity. The stators **1586** and **1587** have grooves **2202** that guide and support the legs **103** and/or the axles **3702**. In certain embodiments, the legs **103** and/or axles **3702** have wheels (item **151** of FIG. **11** or equivalent of other embodiments) that ride on the grooves **2202**.

FIG. **24** shows a side of the stator **1586** opposite to the grooves **2202** of the stator **1586**. In this embodiment, holes **2204** receive the screws that attach the stator spacers **1555** to the stator **1586**. Inner holes **2210** receive the screws that attach the stator **1586** to the flange **4206** of the main shaft **105**. To make some embodiments of the stator **1586** lighter, material is removed from it as shown as cutouts **2206** in this embodiment. For weight considerations as well as clearance of elements of the ball-leg assembly **1670**, the stator **1586** may also include additional cutouts **2208** as in this embodiment.

The embodiments of FIGS. **25**, **26** and **36** will now be referenced to describe one embodiment of an axial force generation mechanism that can be used with the CVT **1500** of FIG. **15**. FIGS. **25** and **26** are partially exploded views. The input shaft **1505** imparts a torque input to the torsion disc **1525**. The torsion disc **1525** couples to a load cam disc **1530** that has ramps **3610**. As the load cam disc **1530** rotates, the ramps **3610** activate the rollers **2504**, which ride up the ramps **3610** of the second load cam disc **1540**. The rollers **2504** then wedge in place, pressed between the ramps of the load cam discs **1530** and **1540**, and transmit both torque and axial force from the load cam disc **1530** to the load cam disc **1540**. In some embodiments, the CVT **1500** includes a roller retainer **1535** to ensure proper alignment of the rollers **2504**. The rollers **2504** may be spherical, cylindrical, barreled, asymmetrical or other shape suitable for a given application. In some embodiments, the rollers **2504** each have individual springs (not shown) attached to the roller retainer **1535** or other structure that bias the rollers **2504** up or down the ramps **3610** as may be desired in some applications. The input disc **1545** in the illustrated embodiment is configured to couple to the load cam disc **1540** and receive both the input torque and the axial force. The axial force then clamps the balls **101** between the input disc **1545**, the output disc **1560**, and the idler **1526**.

In the illustrated embodiment, the load cam disc **1530** is fastened to the torsion disc **1525** with dowel pins. However, other methods of fastening the load cam disc **1530** to the torsion disc **1525** can be used. Moreover, in some embodiments, the load cam disc **1530** is integral with the torsion disc **1525**. In other embodiments, the torsion disc **1525** has the ramps **3610** machined into it to make a single unit for transferring torque and axial force. In the embodiment illustrated, the load cam disc **1540** couples to the input disc **1545** with dowel pins. Again, any other suitable fastening method can be used to couple the input disc **1545** to the load cam disc **1540**. In some embodiments, the input disc **1545** and the load cam disc **1540** are an integral unit, effectively as if the ramps **3610** were built into the input disc **1545**. In yet other embodiments, the axial force generating mechanism may include only one set of ramps **3610**. That is, one of the load cam discs **1530** or **1540** does not have the ramps **3610**, but rather provides a flat surface for contacting the rollers **2504**. Similarly, where the ramps are built into the torsion disc **1525** or the input disc **1545**, one of them may not include the ramps **3610**. In load cam discs **1530**, **1540** in both embodiments having ramps on both or on only one disc, the ramps **3610** and the flat surface on discs without ramps can be formed with a conformal shape conforming to the rollers **2504** surface shape to partially capture the rollers **2504** and to reduce the surface stress levels.

In some embodiments, under certain conditions of operation, a preload axial force to the CVT **1500** is desired. By way of example, at low torque input it is possible for the input disc **1545** to slip on the balls **101**, rather than to achieve frictional traction. In the embodiment illustrated in FIGS. **25** and **26**, axial preload is accomplished in part by coupling a torsion spring **2502** to the torsion disc **1525** and the input disc **1545**. One end of the torsion spring **2502** fits into a hole **2930** (see FIG. **29**) of the torsion disc **1545**, while the other end of the torsion spring **2502** fits into a hole of the input disc **1545**. Of course, the person of ordinary skill in the relevant technology will readily appreciate numerous alternative ways to couple the torsion spring **2502** to the input disc **1545** and the torsion disc **1525**. In other embodiments, the torsion spring **2502** may couple to the roller retainer **1535** and the torsion disc **1525** or the input disc **1545**. In some embodiments where only one of the torsion disc **1525** or input disc **1545** has ramps **3610**, the torsion spring **2502** couples the roller retainer **1535** to the disc with the ramps.

Still referring to the embodiments illustrated in FIGS. **15** **25** and **26**, as mentioned before, in some embodiments the application of axial forces generates reaction forces that are reacted in the CVT **1500**. In this embodiment of the CVT **1500**, a ball thrust bearing aids in managing the reaction forces by transmitting thrust between the hub cap **1460** and the torsion disc **1525**. The thrust bearing has a race **1602** that butts against the hub cap **1460**, which in this embodiment has a recess near its inner bore for receiving the race **1602**. The second race **1603** of the thrust bearing nests in a recess of the torsion disc **1525**. In some embodiments, a wave spring **1515** is incorporated between the race **1602** and the hub **1460** to provide axial preload. In the illustrated embodiment, a bearing **2610** radially supports the hub cap **1460**.

The applicant has discovered that certain configurations of the CVT **1500** are better suited than others to handle a reduction in efficiency of the CVT **1500** due to a phenomenon referred to herein as bearing drag recirculation. This phenomenon arises when a bearing is placed between the torsion disc **1525** and the hub cap **1460** to handle the reaction forces from axial force generation.

In some embodiments as illustrated in FIG. **1**, a needle roller bearing having a diameter about equal to the diameter of the load cam disc **1530** is used to minimize the deflection of the end cap **160**. In underdrive the speed of the torsion disc **157** (input speed) is greater than the speed of the end cap **160** (output speed). In underdrive the needle roller bearing (thrust bearing **163** in that embodiment) generates a drag torque opposite the direction of rotation of the torsion disc **1525**. This drag torque acts on the torsion disc **1525** in the direction counter to the axial loading by the load cam disc **1530**, and acts on the end cap **160** and thus the hub shell **138** and output disc **134** in the direction of the output tending to speed up the rotation of those components, these effects combining to unload the cam loader **154** thereby reduce the amount of axial force in the CVT **1500**. This situation could lead to slip between or among the input disc **110**, balls **101**, and/or output disc **134**.

In overdrive the speed of the torsion disc **1525** is greater than the speed of the end cap **160** and the needle bearing generates a drag torque acting on the torsion disc **1525** in the direction of the rotation of the torsion disc **1525** and acting on the end cap **160** against the output rotation of the end cap **160**. This results in an increase in the axial force being generated in the CVT **1500**. The increase in axial force then causes the system to generate even more drag torque. This feedback phenomenon between axial force and drag torque is what is referred to here as bearing drag recirculation, which ultimately results in reducing the efficiency of the CVT **100**. Additionally, the drag torque acting against the end cap **160** acts as an additional drag on the output of the CVT **100** thereby further reducing its efficiency.

The applicant has discovered various systems and methods for minimizing efficiency losses due to bearing drag recirculation. As shown in FIGS. **25**, **26**, and **40**, instead of using a needle roller bearing configured as described above, some embodiments the CVT **1500** employ a roller thrust bearing having races **1602** and **1603**. Because the amount of drag torque increases with the diameter of the bearing used, the diameter of the races **1602** and **1603** is less than the diameter of the axial force generating load cam disc **1530** and in some embodiments is as small as possible. The diameter of the races **1602** and **1603** could be 10, 20, 30, 40, 50, 60, 70, 80, or 90 percent of the diameter of the load cam disc **1530**. In some embodiments, the diameter of the races **1602** and **1603** is between 30 and 70 percent of the diameter of the load cam disc **1530**. In still other embodiments, the diameter of the races **1602** and **1603** is between 40 and 60 percent of the diameter of the load cam disc **1530**.

When a ball thrust bearing is used, in some embodiments the rollers and/or races are made of ceramic, the races are lubricated and/or superfinished, and/or the number of rollers is minimized while maintaining the desired load capacity. In some embodiments, deep groove radial ball bearings or angular contact bearings may be used. For certain applications, the CVT **1500** may employ magnetic or air bearings as means to minimize bearing drag recirculation. Other approaches to reducing the effects of bearing drag recirculation are discussed below, referencing FIG. **46**, in connection with alternative embodiments of the input shaft **1505** and the main shaft **105**.

FIGS. **27-35** depict examples of certain embodiments of a torque input shaft **1505** and a torsion disc **1525** that can be used with the CVT **1500** of FIG. **15**. The input shaft **1505** and the torsion disc **1525** couple via a splined bore **2710** on the torsion disc **1525** and a splined flange **2720** on the input shaft **1525**. In some embodiments, the input shaft **1505** and the torsion plate **1525** are one piece, made either as a single unit (as illustrated in FIG. **1**) or wherein the input shaft **1505** and the torsion disc **1525** are coupled together by permanent attachment means, such as welding or any other suitable adhesion process. In yet other embodiments, the input shaft **1505** and the torsion disc **1525** are operationally coupled through fasteners such as screws, dowel pins, clips or any other means or method. The particular configuration shown here is preferable in circumstances where it is desired that the input shaft **1505** and the torsion disc **1525** be separate parts, which can handle misalignments and axial displacement due to load cam disc **1530** growth under load, as well as uncouple twisting moments via the splined bore **2710** and the splined shaft **2720**. This configuration is also preferable in certain embodiments because it allows for lower manufacturing tolerances and, consequently, reduced manufacturing costs for a CVT.

Referencing FIGS. **16**, **28-32**, in the illustrated embodiment, the torsion disc **1525** is generally a circular disc having an outer periphery **3110** and a splined inner bore **2710**. One side of the torsion disc **1525** has a recess **3205** that receives the race **1603** of a thrust bearing. The other side of the torsion disc **1525** includes a seat **3210** and a shoulder **3220** for receiving and coupling to the load cam disc **1530**. The torsion disc **1525** includes a raised surface **3230** that rises from the shoulder **3220**, reaches a maximum height in a convex shape, and then falls toward the inner bore **2710**. In one embodiment of the CVT **1500**, the raised surface **3230** partially supports and constrains the torsion spring **2502**, while a set of dowel pins (not shown) helps to retain the torsion spring **2502** in place. In such embodiments, the dowel pins are placed in holes **2920**. The torsion disc **1525** shown here has three splines on its splined bore **2710**. However, in other embodiments the splines can be 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, or more. In some embodiments, the number of splines is 2 to 7, and in others the number of splines is 3, 4, or 5.

In some embodiments, the torsion disc **1525** includes orifices **2910** for receiving dowels that couple the torsion disc **1525** to the load cam disc **1530**. The torsion disc **1525** may also have orifices **2930** for receiving one end of the torsion spring **2502**. In the illustrated embodiment, several orifices **2930** are present in order to accommodate different possible configurations of the torsion spring **2502** as well as to provide for adjustment of preload levels.

The torsion disc **1525** can be of any material of sufficient rigidity and strength to transmit the torques and axial loads expected in a given application. In some embodiments, the material choice is designed to aid in reacting the reaction forces that are generated. For example, hardened steels, steel, aluminum, magnesium, or other metals can be suitable depending on the application while in other applications plastics are suitable.

FIGS. **33-35** show an embodiment of an input torque shaft **1505** for use with the CVT **1500**. The torque input shaft **1505** consists of a hollow, cylindrical body having a splined flange **2720** at one end and a key seat **3310** at the other end. In this embodiment, the key seat **3310** receives a key (not shown) that operationally couples the input shaft **1505** to a freewheel carrier **1510** (see FIGS. **14**, **15**), which itself couples to the freewheel **1420**. The surfaces **2720** and **3410** are shaped to mate with the splined bore **2710** of the torsion disc **1525**. Thus, concave surfaces **2720** of some embodiments will preferably be equal in number to the splines in the splined bore **2710**. In some embodiments, the concave surfaces **2720** may number 1, 2, 3, 4, 5, 6, 7, 8, 9, 10 or more. In some embodiments, the concave surfaces **2720** number 2 to 7, and in others there are 3, 4, or 5 concave surfaces **2720**.

As shown, the input shaft **1505** has several clip grooves that help in retaining various components, such as bearings, spacers, etc., in place axially. The input shaft **1505** is made of a material that can transfer the torques expected in a given application. In some instances, the input shaft **1505** is made of hardened steel, steel, or alloys of other metals while in other embodiments it is made of aluminum, magnesium or any plastic or composite or other suitable material.

FIG. **36** shows an embodiment of a load cam disc **1540** (alternately 1530) that can be used with the CVT **1500**. The disc **1540** is generally a circular ring having a band at its outer periphery. The band is made of ramps **3610**. Some of the ramps **3610** have holes **3620** that receive dowel pins (not shown) for coupling the load cam disc **1530** to the torsion disc **1525** or the load cam disc **1540** to the input disc **1545**. In some embodiments, the ramps **3610** are machined as a single unit with the load cam discs **1530**, **1540**. In other embodiments, the ramps **3610** may be separate from a ring substrate (not shown) and are coupled to it via any known fixation method. In the latter instance, the ramps **3610** and the ring substrate can be made of different materials and by different machining or forging methods. The load cam disc **1540** can be made, for example, of metals or composites.

Referencing FIG. **37** and FIG. **38**, an embodiment of an axle **3702** consists of an elongated cylindrical body having two shoulders **3704** and a waist **3806**. The shoulders **3704** begin at a point beyond the midpoint of the cylindrical body and extend beyond the bore of the ball **101**. The shoulders **3704** of the illustrated embodiment are chamfered, which helps in preventing excessive wear of the bushing **3802** and reduces stress concentration. The ends of the axle **3702** are configured to couple to bearings or other means for interfacing with the legs **103**. In some embodiments, the shoulders **3704** improve assembly of the ball-leg assembly **1670** by providing a support, stop, and/or tolerance reference point for the leg **103**. The waist **3806** in certain embodiments serves as an oil reservoir. In this embodiment, a bushing **3802** envelops the axle **3702** inside the bore of the ball **101**. In other embodiments, bearings are used instead of the bushing **3802**. In those embodiments, the waist **3806** ends where the bearings fit inside the ball **101**. The bearings can be roller bearings, drawn cup needle rollers, caged needle rollers, journal bearings, or bushings. In some embodiments, it is preferred that the bearings are caged needle bearings or other retained bearings. In attempting to utilize general friction bearings, the CVT **100**, **1500** often fails or seizes due to a migration of the bearings or rolling elements of the bearings along the axles **3702**, **102** out of the balls **101** to a point where they interfere with the legs **103** and seize the balls **101**. It is believed that this migration is caused by force or strain waves distributed through the balls **101** during operation. Extensive testing and design has lead to this understanding and the Applicant's believe that the use of caged needle rollers or other retained bearings significantly and unexpectedly lead to longer life and improved durability of certain embodiments of the CVT **100**, **1500**. Embodiments utilizing bushings and journal material also aid in the reduction of failures due to this phenomenon. The bushing **3802** can be replaced by, for example, a babbitt lining that coats either or both of the ball **101** or axle **3702**. In yet other embodiments, the axle **3702** is made of bronze and provides a bearing surface for the ball **101** without the need for bearings, bushing, or other linings. In some embodiments, the ball **101** is supported by caged needle bearings separated by a spacer (not shown) located in the middle portion of the bore of the ball **101**. Additionally, in other embodiments, spacers mount on the shoulders **3704** and separate the caged needle bearings from components of the leg **103**. The axle **3702** can be made of steel, aluminum, magnesium, bronze, or any other metal or alloy. In certain embodiments, the axle **3702** is made of plastic or ceramic materials.

One embodiment of the main shaft **105** is depicted in FIGS. **41-43**. The main shaft **105** is an elongated body having an inner bore **4305** for receiving a shift rod **112** (see FIGS. **16** and **40**). As implemented in the CVT **1500**, the main shaft **105** is a single piece axle that provides support for many of the components of the CVT **1500**. In embodiments where a single piece axle is utilized for the main shaft **105**, the main shaft **105** reduces or eliminates tolerance stacks in certain embodiments of the CVT **1500**. Furthermore, as compared with multiple piece axles, the single piece main shaft **105** provides greater rigidity and stability to the CVT **1500**.

The main shaft **105** also includes a through slot **4204** that receives and allows the shift pin **114** to move axially, that is, along the longitudinal axis of the main shaft **105**. The size of the slots **4204** can be chosen to provide shift stops for selectively determining a ratio range for a given application of the CVT **1500**. For example, a CVT **1500** can be configured to have a greater underdrive range than overdrive range, or vice-versa, by choosing the appropriate dimension and/or location of the slots **4204**. By way of example, if the slot **4204** shown in FIG. **42** is assumed to provide for the full shift range that the CVT **1500** is capable of, a slot shorter than the slot **4204** would reduce the ratio range. If the slot **4204** were to be shortened on the right side of FIG. **42**, the underdrive range would be reduced. Conversely, if the slot **4204** were to be shortened on the left side of FIG. **42**, the overdrive range would be reduced.

In this embodiment, a flange **4206** and a shoulder **4208** extend from the main shaft **105** in the radial direction. As already described, the flange **4206** and the shoulder **4208** facilitate the fixation of the stator **1586** to the main shaft **105**. In some embodiments, the bore of the stator **1586** is sized to mount to the main shaft **105** such that the shoulder **4208** can be dispensed with. In other embodiments, the shoulder **4208** and/or the flange **4206** can be a separate part from the main shaft **105**. In those instances, the shoulder **4208** and/or flange **4206** mount coaxially about the main shaft **105** and affix to it by any well known means in the relevant technology. In the embodiment depicted, the main shaft **105** includes a key seat **4202** for receiving a key **1606** that rotationally fixes the flange **1610** (see FIG. **16**). The key **1606** may be a woodruff key. The main shaft **105** of some embodiments is made of a metal suitable in terms of manufacturability, cost, strength, and rigidity. For example, the main shaft can be made of steel, magnesium, aluminum or other metals or alloys.

The operation of the hub **1400** having one embodiment of the CVT **1500** described above will now be described with particular reference to FIGS. **39** and **40**. The freewheel **1420** receives torque from a bicycle chain (not shown). Since the freewheel **1420** is fixed to the freewheel carrier **1510**, the freewheel **1420** imparts the torque to the freewheel carrier **1510**, which in turns transmits the torque to the input shaft **1505** via a key coupling (not shown). The input shaft **1505**, riding on needle bearings **4010** and **4020** mounted on the main shaft **105**, inputs the torque to the torsion disc **1525** via the splined bore **2710** and splined surfaces **2720** and **3410** of the input shaft **1505**. Needle bearing **4010** is preferably placed near or underneath the freewheel carrier **1510** and/or freewheel **1420**. This placement provides appropriate support to the input shaft **1505** to prevent transmission of radial loading from the freewheel carrier **1510** as a bending load through the CVT **1400**. Additionally, in some embodiments a spacer **4030** is provided between the needle bearings **4010** and **4020**. The spacer **4030** may be made of, for example, Teflon.

As the torsion disc **1525** rotates, the load cam disc **1530** coupled to the torsion disc **1525** follows the rotation and, consequently, the ramps **3610** energize the rollers **2504**. The rollers **2504** ride up the ramps **3610** of the load cam disc **1540** and become wedged between the load cam disc **1530** and the load cam disc **1540**. The wedging of the rollers **2504** results in a transfer of both torque and axial force from the load cam disc **1530** to the load cam disc **1540**. The roller cage **1535** serves to retain the rollers **2504** in proper alignment.

Because the load cam disc **1540** is rigidly coupled to the input disc **1545**, the load cam disc **1540** transfers both axial force and torque to the input disc **1545**, which then imparts the axial force and torque to the balls **101** via frictional contact. As the input disc **1545** rotates under the torque it receives from the load cam disc **1540**, the frictional contact between the input disc **1545** and the balls **101** forces the balls **101** to spin about the axles **3702**. In this embodiment, the axles **3702** are constrained from rotating with the balls **101** about their own longitudinal axis; however, the axles **3702** can pivot or tilt about the center of the balls **101**, as in during shifting.

The input disc **1545**, output disc **1560**, and idler **1526** are in frictional contact with the balls **101**. As the balls **101** spin on the axles **3702**, the balls **101** impart a torque to the output disc **1560**, forcing the output disc **1560** to rotate about the shaft **105**. Because the output disc **1560** is coupled rigidly to the hub shell **138**, the output disc **1560** imparts the output torque to the hub shell **138**. The hub shell **138** is mounted coaxially and rotatably about the main shaft **105**. The hub shell **138** then transmits the output torque to the wheel of the bicycle via well known methods such as spokes.

Still referring to FIGS. **39** and **40**, shifting of the ratio of input speed to output speed, and consequently a shift in the ratio of input torque to output torque, is accomplished by tilting the rotational axis of the balls **101**, which requires actuating a shift in the angle of the axles **3702**. A shift in the transmission ratio involves actuating an axial movement of the shift rod **112** in the main shaft **105**, or in rotation of the shift rod **312** of FIG. **3**. The shift rod **112** translates axially the pin **114**, which is in contact with the shift cams **1527** via the bore **1910** in the extension **1528**. The axial movement of the shift pin **114** causes a corresponding axial movement of the shift cams **1527**. Because the shift cams **1527** engage the legs **103** (via cam wheels **152**, for example), the legs **103** move radially as the legs **103** move along the shift cam profile **2110**. Since the legs **103** are connected to the axles **3702**, the legs **103** act as levers that pivot the axles **3702** about the center of the balls **101**. The pivoting of the axles **3702** causes the balls **101** to change axis of rotation and, consequently, produce a ratio shift in the transmission.

FIG. **44** and FIG. **45** show an embodiment of a CVT **4400** having an axial force generating mechanism that includes one load cam disc **4440** acting on the input disc **1545** and another load cam disc **4420** acting on the output disc **1560**. In this embodiment, the load cam discs **4440** and **4420** incorporate ramps such as ramps **3610** of the load cam discs **1530** and **1540**. In this embodiment, neither of the input disc **1545** or the output disc **1560** has ramps or is coupled to discs with ramps. However, in other embodiments, it may be desirable to provide one or both of the input disc **1545** or output disc **1560** with discs having ramps, or building the ramps into the input disc **1545** and/or output disc **1560** to cooperate with the load cam discs **4420**, **4440**. The CVT **4400** of some embodiments further includes a roller retainer **4430** to house and align a set of rollers (not shown) that is between the load cam disc **4420** and the output disc **1560**. In the embodiment shown, the roller retainer **4430** radially pilots on the output disc **1560**. Similarly, there is a roller retainer **4410** between the load cam disc **4440** and the input disc **1545**. The rollers and discs described with reference to these embodiments can be of any type or shape as described above for previous axial force generating devices. In some embodiments the angles of the ramps incline from the surface of the disc at an angle that is (or is between) 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15 degrees or more or any portion between any of these.

FIG. **46** illustrates an embodiment of a CVT **1600** having an input shaft **4605** and a main shaft **4625** adapted to decrease bearing drag recirculation effects. The CVT **100** includes an axial force generator **165** which generates an axial force that is reacted in part by a needle roller bearing **4620**. A hub cap **4660** reacts drag torque and axial forces from the needle roller bearing **4620**. In other embodiments, the needle roller bearing **4620** is replaced by a ball thrust bearing and in other embodiments the ball thrust bearing has a diameter smaller than the diameter of the needle roller bearing **4620**.

In this embodiment, the main shaft **4625** has a shoulder **4650** that provides a reaction surface for a washer **4615**, which can also be a clip, for example (all of which are integral in some embodiments). The input shaft **4605** is fitted with an extension **1410** that reacts against a bearing **4645**. The bearing **4645** can be a thrust bearing. As shown, the input shaft **4605** and driver disc (similar to the torsion disc **1525**) are a single piece. However, in other embodiments the input shaft **4605** may be coupled to a torsion disc **1525**, for example, by threading, keying, or other fastening means. In the illustrated embodiment, some of the reaction force arising from the generation of axial force is reacted to the main shaft **4625**, thereby reducing bearing drag recirculation. In yet another embodiment (not shown), the extension **1410** is reacted against angular thrust bearings that also support the input shaft **4605** on the main shaft **4625**. In this latter embodiment, the shoulder **4650** and washer **4615** are not required. Rather, the main shaft **4625** would be adapted to support and retain the angular thrust bearings.

In many embodiments described herein, lubricating fluids are utilized to reduce friction of the bearings supporting many of the elements described. Furthermore, some embodiments benefit from fluids that provide a higher coefficient of traction to the traction components transmitting torque through the transmissions. Such fluids, referred to as "traction fluids" suitable for use in certain embodiments include commercially available Santotrac 50, 5CST AF from Ashland oil, OS#155378 from Lubrizol, IVT Fluid #SL-2003B21-A from Exxon Mobile as well as any other suitable lubricant. In some embodiments the traction fluid for the torque transmitting components is separate from the lubricant that lubricates the bearings.

The foregoing description details certain embodiments of the invention. It will be appreciated, however, that no matter how detailed the foregoing appears in text, the invention can be practiced in many ways. As is also stated above, it should be noted that the use of particular terminology when describing certain features or aspects of the invention should not be taken to imply that the terminology is being re-defined herein to be restricted to including any specific characteristics of the features or aspects of the invention with which that terminology is associated. The scope of the invention should therefore be construed in accordance with the appended claims and any equivalents thereof.

What is claimed is:

1. A system comprising:

a continuously variable transmission (CVT) comprising:

an input disc arranged along a longitudinal axis of the CVT,

an output disc arranged along the longitudinal axis of the CVT,

a plurality of balls arranged between, and in contact with, the input disc and the output disc, each ball provided with an adjustable tiltable axis of rotation, and

an idler located radially inward of the balls, the idler arranged between the input disc and the output disc, each ball in contact with the idler; and

an input torque system comprising:

an input shaft having a splined flange,

a torsion disc having a splined bore, and

a spring coupled to the torsion disc, wherein the torsion disc comprises a plurality of holes adapted to receive one end of the spring, the torsion disc configured to fasten one end of the spring so as to facilitate the adjustment of an axial pre-load between the balls, the input disc, the output disc, and the idler, and wherein the splined flange and splined bore are adapted to couple together operationally to transmit torque from the input shaft to the torsion disc.

2. The system of claim **1**, wherein the splined flange comprises at least two splines.

3. The system of claim **2**, wherein the splined flange comprises a key seat for receiving a key adapted to transfer torque to the input shaft.

4. The system of claim **1**, wherein the splined bore comprises at least two splines.

5. The system of claim **1**, wherein the torsion disc comprises a recess for receiving a bearing race.

6. The system of claim **5**, wherein the bearing race is integral with the torsion disc.

7. The system of claim **1**, further comprising a first load cam disc coupled to the torsion disc.

8. The system of claim **7**, further comprising a second load cam disc adapted to couple to the input disc.

9. The system of claim **8**, further comprising rollers between the first load cam disc and the second load cam disc.

10. The system of claim **9**, wherein the rollers are spherical.

11. The system of claim **9**, wherein the rollers are cylindrical.

12. The system of claim **9**, wherein the first load cam disc comprises ramps.

13. The system of claim **9**, wherein the second load cam disc comprises ramps.

14. A system comprising:

a continuously variable transmission (CVT) comprising

an input disc arranged along a longitudinal axis of the CVT,

an output disc arranged along the longitudinal axis of the CVT,

a plurality of balls arranged between, and in contact with, the input disc and the output disc, each ball provided with an adjustable tiltable axis of rotation, and

an idler located radially inward of the balls, the idler arranged between the input disc and the output disc, each ball in contact with the idler;

an input torque system comprising

an input shaft having a splined flange,

a torsion disc having a splined bore,

a spring coupled to the torsion disc,

a first load cam disc coupled to the torsion disc,

a second load cam disc adapted to couple to the input disc, and

rollers between the first load cam disc and the second load cam disc, wherein the torsion disc is provided with a plurality of holes adapted to receive one end of the spring, the torsion disc configured to fasten one end of the spring so as to facilitate the adjustment of an axial pre-load between the first load cam disc and the second load cam disc, and wherein the splined flange and the splined bore are adapted to couple together operationally to transmit torque from the input shaft to the torsion disc.

15. The system of claim **14**, wherein the first load cam disc comprises ramps.

16. The system of claim **14**, wherein the second load cam disc comprises ramps.

17. The system of claim **14**, wherein the torsion disc comprises a recess for receiving a bearing race and wherein the bearing race is integral with the torsion disc.

* * * * *

UNITED STATES PATENT AND TRADEMARK OFFICE

CERTIFICATE OF CORRECTION

PATENT NO. : 8,920,285 B2
APPLICATION NO. : 13/710304
DATED : December 30, 2014
INVENTOR(S) : Smithson et al.

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

Title Page

Page 1 (item 72, Inventors) at line 7, change “Mathew P Simister” to --Matthew P Simister--.

Signed and Sealed this
Tenth Day of May, 2016

Michelle K. Lee

Michelle K. Lee
Director of the United States Patent and Trademark Office

UNITED STATES PATENT AND TRADEMARK OFFICE

CERTIFICATE OF CORRECTION

PATENT NO.	: 8,920,285 B2
APPLICATION NO.	: 13/710304
DATED	: December 30, 2014
INVENTOR(S)	: Robert A. Smithson et al.

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

A petition under 1.182 is required to correct the spelling of the inventor's name. The Certificate of Correction which issued on May 10, 2016 was published in error and should not have been issued for this patent. The Certificate of Correction issued on May 10, 2016 is vacated.

Signed and Sealed this
Third Day of January, 2017

Michelle K. Lee

Michelle K. Lee
Director of the United States Patent and Trademark Office